

287068

N70 72159

APS-5200-R

DESIGN AND FABRICATION OF A HIGH- PERFORMANCE BRAYTON CYCLE RADIAL-FLOW GAS GENERATOR

Prepared under Contract No. NAS 3-2778 by
AIRESEARCH MFG. CO. of ARIZONA
a division of THE GARRETT CORP.
Phoenix, Arizona

Reproduced by
NATIONAL TECHNICAL
INFORMATION SERVICE
US Department of Commerce
Springfield, VA 22151

FACILITY FORM 602	N70-72159	
	(ACCESSION NUMBER)	(THRU)
	39	NONE
	(PAGES)	(CODE)
CRH 8853		
(NASA CR OR TMX OR AD NUMBER)		
(CATEGORY)		

38

DESIGN AND FABRICATION OF
A HIGH-PERFORMANCE BRAYTON CYCLE
RADIAL-FLOW GAS GENERATOR

APS-5200-R

June 13, 1966

Prepared by Engineering Staff

Initial Issue
Approved by

H. H. Milligan
H. H. Milligan, Advanced Power
Systems Project



AIRESEARCH MANUFACTURING COMPANY

A DIVISION OF THE GARRETT CORPORATION
PHOENIX, ARIZONA

CONTRACTOR REPORT

DESIGN AND FABRICATION OF A HIGH-

PERFORMANCE BRAYTON CYCLE

RADIAL-FLOW GAS GENERATOR

Prepared under Contract NAS 3-2778 by
AiResearch Manufacturing Company of Arizona
Phoenix, Arizona

This report, reproduced photographically from copy supplied
by the contractor, was originally issued as AiResearch En-
gineering Report APS-5200-R, June 13, 1966.

Technical Management
Space Power Systems Division
Jack A. Heller

Fluid System Components Division
George K. Fischer
Harold E. Rohlik

Lewis Research Center

Cleveland, Ohio

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

ia



TABLE OF CONTENTS

<u>Title</u>	<u>Page</u>
SUMMARY	1
1.0 INTRODUCTION	3
2.0 SELECTION OF DESIGN SPEED	5
3.0 GAS GENERATOR DESIGN	11
3.1 Gas Generator Description	11
3.2 Compressor and Turbine	11
3.3 Dynamic Analysis	16
3.4 Journal Gas Bearings	22
3.4.1 Design Considerations and Bearing Selection	22
3.4.2 Gas Bearing Analysis	27
3.4.3 Gas Bearing Testing	29
3.4.4 Gas Generator Journal Bearing Configuration	36
3.5 Thrust Gas Bearings	38
3.5.1 Thrust Bearing Design	38
3.5.2 Thrust Bearing Tests	47
3.6 Thermal Analysis	49
3.7 Shock and Vibration Analysis	61
3.8 Instrumentation	67
3.8.1 Capacitance Probes	68
3.8.2 Strain Gauges	71
3.8.3 Thermocouples	74
3.8.4 Additional Instrumentation	74



TABLE OF CONTENTS (Contd.)

<u>Title</u>	Page
4.0 FINAL GAS GENERATOR CONFIGURATION	83
4.1 Gas Generator Description	83
4.1.1 Rotating Assembly	83
4.1.2 Compressor/Turbine Journal Bearing Assembly	83
4.1.3 Gimbal and Thrust Bearing Assembly	87
4.1.4 Compressor Scroll and Diffuser Assembly	88
4.1.5 Turbine Scroll and Nozzle Assembly	88
4.1.6 Main Frame Assembly	89
4.1.7 Mount Assembly	89
4.2 Component Configurations and Fabrication	90
4.3 Inspection	90
5.0 ACCEPTANCE TEST	112
5.1 Test Installation	112
5.2 Assembly of Gas Generator, First Shipping Unit	116
5.3 Initial Test and Calibration of First Shipping Unit	117
5.4 Acceptance Test of First Shipping Unit	118
APPENDIX I - Glossary	126



LIST OF FIGURES

<u>Figure No.</u>	<u>Title</u>	<u>Page</u>
1	NASA Brayton-Cycle Power System	6
2	Compressor Design Point Study Results	9
3	Turbine Design Point Study Results	10
4	NASA Brayton-Cycle Radial Gas Generator	12
5	NASA Compressor Research Package	13
6	NASA Turbine Research Package	14
7	6 in. Diameter Compressor Test Results	15
8	Gas Generator Configurations	17
9	Critical Speed vs Bearing Spring Rate for NASA Gas Generator, Initial Results	19
10	Bearing Load vs Rotor Speed NASA Gas Generator Final Configuration	20
11	Bearing Mounting Methods	25
12	Gas Bearing Shoe Mounting Methods	26
13	Single Shoe Test Rig	31
14	Dynamic Two-Bearing Test Rig	32
15	Gas Bearing Test Facility Instrumentation	33
16	Predicted Gas Generator Journal Bearing Performance	39
17	Thrust Bearing and Gimbal Assembly	41
18	Hydrostatic Thrust Bearing for NASA Gas Generator	42
19	Hydrodynamic Thrust Bearing for NASA Gas Generator	43
20	Hydrostatic Thrust Bearing Load Capacity	44



LIST OF FIGURES (Contd.)

<u>Figure No.</u>	<u>Title</u>	<u>Page</u>
21	Hydrodynamic Thrust Bearing, Hydrostatic Load Capacity	45
22	NASA Gas Generator Hydrodynamic Thrust Bearing Performance	46
23	Company-Sponsored Bearing Test Rig	48
24	Steady State Temperature Distribution	51
25	Heat Flow Diagram	52
26	Shaft Temperatures at Different Values of Time	53
27	Transient and Steady-State Compressor and Turbine Shroud Clearance	54
28	Compressor and Turbine Bearing and Shaft Temperatures at Different Values of Time	55
29	Transient and Steady-State Bearing Journal to Carrier Relative Expansion	57
30	Transient and Steady-State of the Mass Center Line of the Shaft	58
31	Transient and Steady-State Journal Bearing Loads	59
32	Transient and Steady-State Temperatures of Representative Points in the Gas Generator	60
33	Steady-State Shaft Temperatures for Various Cooling Schemes	62
34	Schematic of Gas Generator Frame Mount Points (Body 2)	64
35	Schematic of Gas Generator Shaft Mount Points (Body 3)	64
36	Load, Shear, and Bending Moment Diagrams for Rectangular Shock Pulse in Z Direction During Launch Condition	65



LIST OF FIGURES (Contd.)

<u>Figure No.</u>	<u>Title</u>	<u>Page</u>
37	Test Results on Proximity Probe Calibrations	69
38	Shaft Capacitance Probe	70
39	Predicted and Actual Test Results of Effect of Curved Surface on Flat-Plate Calibration of Wayne-Kerr Proximity Probe System	72
40	Proximity Probe and Calibration Fixture	73
41	Representative Strain-Gauge Circuit	75
42	Magnetic Speed Pickup	76
43	Instrumentation Installation Partial Assembly NASA Gas Generator	77
44	Instrumentation Installation, Final Wiring NASA Gas Generator	78
45	Instrumentation Installation, Final Wiring NASA Gas Generator	79
46	Instrumentation Installation, Final Wiring NASA Gas Generator	80
47	Instrumentation Receptacle Plate	81
48	NASA Brayton-Cycle Radial Gas Generator	84
49	NASA Brayton-Cycle Radial Gas Generator with Materials of Pertinent Parts Noted	91
50	NASA Gas Generator Part 699300, Component Disassembly	92
51	NASA Gas Generator Part 699300, Component Disassembly	93
52	Bearing Carrier and Frame Assembly	94



LIST OF FIGURES (Contd.)

<u>Figure No.</u>	<u>Title</u>	<u>Page</u>
53	Rotating Component Group Assembly	95
54	Shaft Assembly and Tension Bolt	96
55	Turbine Scroll and Nozzle Assembly	97
56	Compressor Diffuser Assembly	98
57	Compressor Impeller	99
58	Compressor Scroll Assembly	100
59	Shoe and Diaphragm Assembly, Matched Set	101
60	Shoe and Solid Mount, Matched Set	102
61	Hydrostatic Thrust Bearing Stator	103
62	Hydrodynamic Thrust Bearing Stator	104
63	Turbine and Compressor Bearing Carriers	105
64	Outer Housing Assembly	106
65	NASA Brayton-Cycle Gas Generator	107
66	Critical Part Inspection and Serialization Records for Gas Generator P-A	108
67	Critical Part Inspection and Serialization Records for Gas Generator P-B	109
68	Assembly Inspection and Laboratory Traveler, Gas Generator P-A	110
69	Gas Generator Acceptance Test Installation Schematic	113



LIST OF FIGURES (Contd.)

<u>Figure No.</u>	<u>Title</u>	<u>Page</u>
70	Acceptance Test Mock-Up of Gas Generator Installed in Test Cell	114
71	Acceptance Test Mock-Up of Gas Generator Installed in Test Cell	115
72	Wayne-Kerr Proximity Meters, NASA Gas Generator Acceptance Test	119
73	Gas Bearing Control Panel and Instrumentation NASA Gas Generator Acceptance Test	120
74	Qualification Test Log	122
75	NASA Gas Generator Acceptance Test Log Sheet (Sheet 1a)	123
76	NASA Gas Generator Acceptance Test Log Sheet (Sheet 1b)	124
77	NASA Gas Generator Acceptance Test Tape and Oscillograph Log Sheets	125

LIST OF TABLES

<u>Table No.</u>	<u>Title</u>	<u>Page</u>
1	Design Conditions Fixed by Contract NAS3-2778	7
2	Summary of Spring Rates for Launch Conditions	65
3	Summary of Maximum Forces at Mounts for Rec- tangular Shock Pulse in X Direction During Launch Condition	65
4	Summary of Maximum Forces at Mounts for Rec- tangular Shock Pulse in Z Direction During Launch Condition	65
5	NASA Brayton-Cycle Gas Generator Parts List	85



AIRESEARCH MANUFACTURING COMPANY
A DIVISION OF THE GARRETT CORPORATION
PHOENIX, ARIZONA

CONTRIBUTING PERSONNEL

The following personnel of the AiResearch Manufacturing Company of Arizona were responsible for the conduct of the work performed under Contract NAS3-2778 and the contents of this report.

D. J. Brandes
R. O. Bullock
R. W. Elliott
R. B. Gildersleeve
F. E. Johnson
L. A. Matsch
H. H. Milligan
E. A. Mock
L. W. Norman
G. L. Perrone
E. L. Wheeler



AIRESEARCH MANUFACTURING COMPANY OF ARIZONA
A DIVISION OF THE GARRETT CORPORATION
PHOENIX ARIZONA

DESIGN AND FABRICATION OF A HIGH-PERFORMANCE
BRAYTON CYCLE RADIAL-FLOW GAS GENERATOR,
AIRESEARCH MANUFACTURING COMPANY OF ARIZONA

SUMMARY

The NASA-Lewis Research Center is presently engaged in an investigation of Brayton-cycle space-power systems that use solar or nuclear energy as the heat source and an inert gas as the working fluid. The turbomachinery required for the system and component investigations was defined by the NASA for both radial and axial-flow machinery. Under Contract NAS3-2778, three pieces of radial turbomachinery were designed and fabricated, as follows:

Compressor and Turbine Research Packages - The two compressor and turbine research packages each include a high-performance single-stage radial wheel and attendant components and a suitable set of running gear with oil-lubricated bearings. Both research packages are to be used to evaluate component aerodynamic performance.

Gas Generator - This unit combines the turbine and compressor of the two research packages into a single hot unit, with the running gear including gas-lubricated bearings. The gas generator will be used to evaluate the Brayton-cycle radial turbomachinery in a ground test loop.



This report describes the selection of the gas generator design speed and the design, fabrication, inspection, instrumentation, and testing of the gas generator prior to delivery to the NASA. Shown below is the specified design point of the turbine and compressor as individual components and as included in the radial turbomachinery gas generator:

	<u>Turbine</u>	<u>Compressor</u>	<u>Gas Generator</u>
Working fluid	Argon	Argon	Argon
Flow rate, lbs per sec.	1.184	0.621	0.611
Turbine inlet temperature, °R	520	-	1950
Turbine inlet pressure, psia	13.2	-	13.2
Turbine total pressure ratio	1.56	-	1.56
Compressor inlet temperature, °R	-	520	536
Compressor inlet pressure, psia	-	6.0	6.0
Compressor pressure ratio	-	2.30	2.30

Two cold research packages each of the turbine and compressor were delivered to the NASA for aerodynamic performance investigations. Two gas generators were fabricated and also delivered to the NASA; one as a complete unit and the second as separate parts. Successful dynamic and mechanical integration of the thrust and journal gas bearings with the gas generator was accomplished. Prior to shipment to the NASA, the assembled unit was subjected to an Acceptance Test which demonstrated satisfactory performance on air at approximately 500°F turbine inlet temperature.



1.0 INTRODUCTION

The NASA-Lewis Research Center is presently conducting an evaluation of closed, recuperated Brayton-cycle space power systems that use solar or nuclear energy as the heat source and an inert gas as the working fluid. Both axial- and radial-flow types of turbomachinery will be evaluated. This contract (NAS3-2778) calls for the design and fabrication of the following radial-flow turbomachinery:

Compressor and Turbine Research Packages - The two compressor and turbine research packages each include a high-performance single-stage radial wheel and attendant components and a suitable set of running gear with oil-lubricated bearings. Both research packages are to be used to evaluate component aerodynamic performance.

Gas Generator - This unit combines the turbine and compressor of the two research packages into a single hot unit, with the running gear including gas-lubricated bearings. The gas generator will be used to evaluate the Brayton-cycle radial turbomachinery in a ground test loop.

The gas generator is but one of the components utilized in the closed Brayton-cycle space power system. However, it is perhaps the most critical from a performance and reliability standpoint. The gas generator consists of a single-stage radial compressor utilized for the compression of the inert working fluid (argon) mounted on a shaft supported by gas journal and thrust bearings, driven by a single-stage, radial-inflow power turbine (mounted on the same shaft). The gas, partially expanded, is exhausted from the gas generator compressor drive turbine into a second-stage free turbine which drives an alternator on the same shaft. This latter package is separate from the gas generator unit.



AIRESEARCH MANUFACTURING COMPANY OF ARIZONA
A DIVISION OF THE GARRETT CORPORATION
PHOENIX, ARIZONA

The contract effort, performed to design, fabricate, and acceptance test the radial-flow gas generator, is summarized in Section 3.0 of this report. The program included the scaling of existing AiResearch turbine and compressor designs, and development of the mechanical and dynamic integration of the gas bearings into the gas generator.

The significance of the program is the integration of gas bearings with turbomachinery that culminates as a gas-bearing gas generator designed for operation on inert gas in a zero-g environment--a first of its kind. Gas bearings appear to be a promising solution to the bearing requirements of long-life Brayton-cycle space power turbomachinery.

The guidance and assistance given to this contractor by Messrs. J. A. Heller, H. E. Rohlik, and G. K. Fischer of NASA-Lewis are acknowledged. The technical assistance and consultation provided by The Franklin Institute (Dr. E. J. Gunter, Jr.), and by Dr. V. Castelli, in the design of gas journal bearings is also acknowledged.



2.0 SELECTION OF DESIGN SPEED

Figure 1 shows a schematic of the NASA Brayton-cycle space power system. Contract NAS3-2778 calls for the design and fabrication of a turbine research test package, a compressor research test package, and a gas generator with the same aerodynamic components as were used in the test packages. As specified by the contract, the gas generator and research test packages have identical design conditions when corrected mass-flow rates are compared. Table 1 presents a summary of the design conditions as specified by the contract. In addition to the conditions listed in Table 1, the most important remaining turbomachinery variables are:

- (a) Shaft speed, N_1
- (b) Compressor specific speed, N_{S_c}

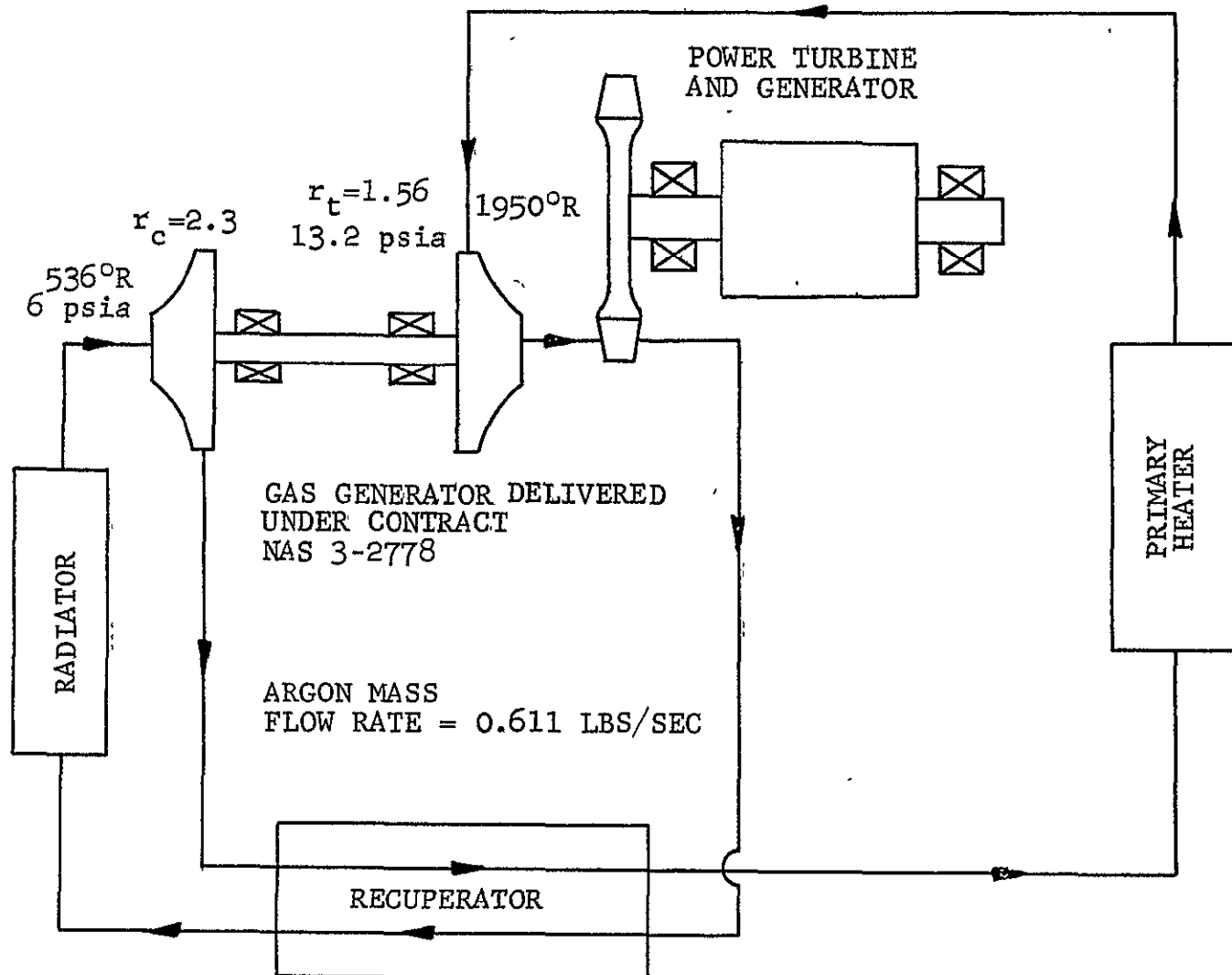
From the expression of compressor specific speed

$$N_{S_c} = \frac{N_1}{60} \left(\frac{1}{R_1 T_1} \right)^{1/4} \left(\frac{\dot{m}}{P_1} \right)^{1/2} \left[\frac{(\gamma - 1)M}{\gamma g(r_c^{\theta_1} - 1)} \right]^{3/4}$$

it can be seen that the compressor specific speed is a function only of shaft speed, since the remaining variables are fixed by the contract. (A list of the symbols used throughout this report can be found in the Glossary, Appendix I.) Therefore, the shaft speed is the only remaining variable to be specified.



AIRESEARCH MANUFACTURING COMPANY
A DIVISION OF THE GARRETT CORPORATION
PHOENIX, ARIZONA



NASA BRAYTON-CYCLE-POWER SYSTEM

A13896

FIGURE 1



AIRESEARCH MANUFACTURING COMPANY OF ARIZONA
A DIVISION OF THE GARRETT CORPORATION
PHOENIX, ARIZONA

TABLE 1

DESIGN PARAMETERS FIXED BY
CONTRACT NAS 3-2778

	<u>Turbine Package</u>	<u>Compressor Package</u>	<u>Gas Generator</u>
Working fluid	Argon	Argon	Argon
Mass flow rate, W - lbs per sec.	1.184	0.621	0.611
Turbine inlet temperature, T_3 , $^{\circ}R$	520	-	1950
Turbine inlet pressure, P_3 , psia	13.2	-	13.2
Turbine pressure ratio, r_{t1}	1.56	-	1.56
Compressor inlet temperature, T_1 , $^{\circ}R$	-	520	536
Compressor inlet pressure, P_1 , psia	-	6.0	6.0
Compressor pressure ratio, r_c	-	2.30	2.30
Corrected mass flow rate:			
$W \sqrt{\theta}/\delta$ turbine, lbs per sec.	1.318		1.318
$W \sqrt{\theta}/\delta$ compressor, lbs per sec.		1.521	1.521



With the design-point conditions listed in Table 1, a recuperator effectiveness (E_R) of 0.85, and a pressure-loss parameter (β) of 0.90, a gas generator design-point study was conducted to establish the gas generator thermodynamic and aerodynamic operating conditions. Figures 2 and 3 illustrate the variation of wheel diameters, component and cycle efficiencies, component specific speeds, and turbine pressure ratio over a range of shaft speeds with the turbine and compressor matched. As noted on the figures, a shaft speed of 38,500 rpm was selected as discussed below.

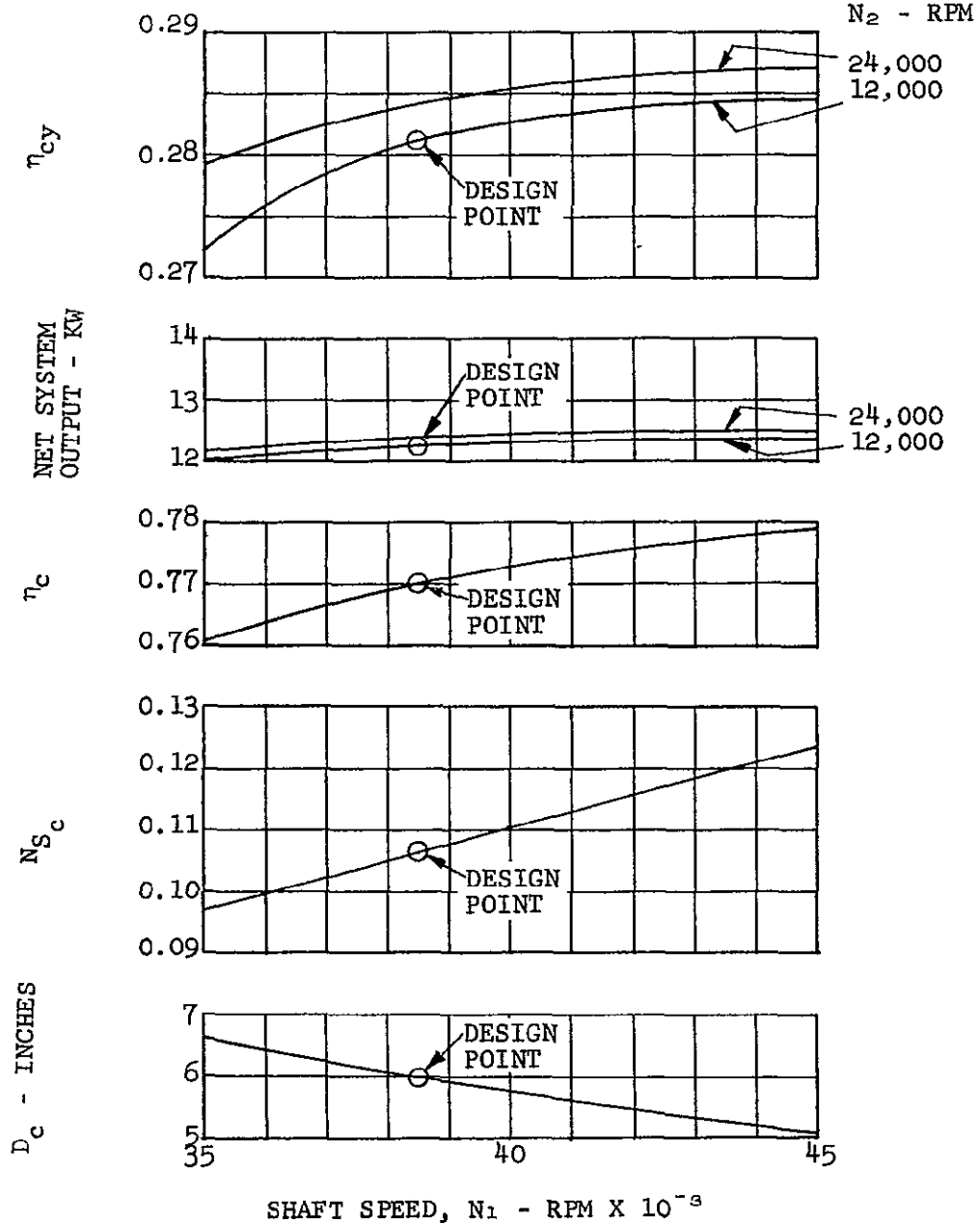
- (a) The compressor specific speed of 0.106, corresponding to a shaft speed of 38,500 rpm, makes AiResearch compressor experience with similar specific speeds directly applicable. A more favorable specific speed of 0.09, based on a system weight optimization to reduce the weight 5 percent, would result in a speed of 33,000 rpm, which is undesirable, as discussed in (c) below.
- (b) Reasonable adiabatic efficiencies occur for the compressor and turbines at a shaft speed of 38,500 rpm, whereas at lower speeds the component efficiency and, thus, the cycle efficiency drop off rapidly.
- (c) Turbine and compressor wheel diameters of 6 inches occur at a shaft speed of 38,500 rpm with reasonably obtainable adiabatic efficiencies. An increase in the shaft speed would reduce the wheel size and result in more restrictive manufacturing tolerances necessary to maintain hydraulically smooth aerodynamic passages.



COMPRESSOR INLET TEMPERATURE, $T_1 = 536^{\circ}\text{R}$
TURBINE INLET TEMPERATURE, $T_3 = 1950^{\circ}\text{R}$
RECUPERATOR EFFECTIVENESS, $E_R = 0.85$
WORKING FLUID = ARGON

TURBINE PRESSURE RATIO
COMPRESSOR PRESSURE RATIO, $\beta = 0.90$

COMPRESSOR PRESSURE RATIO, $r_c = 2.30$
COMPRESSOR INLET PRESSURE, $P_1^c = 6.0 \text{ PSIA}$
MASS FLOW RATE, $\dot{m} = 0.611 \text{ LBS PER SEC}$



COMPRESSOR DESIGN POINT

STUDY RESULTS.

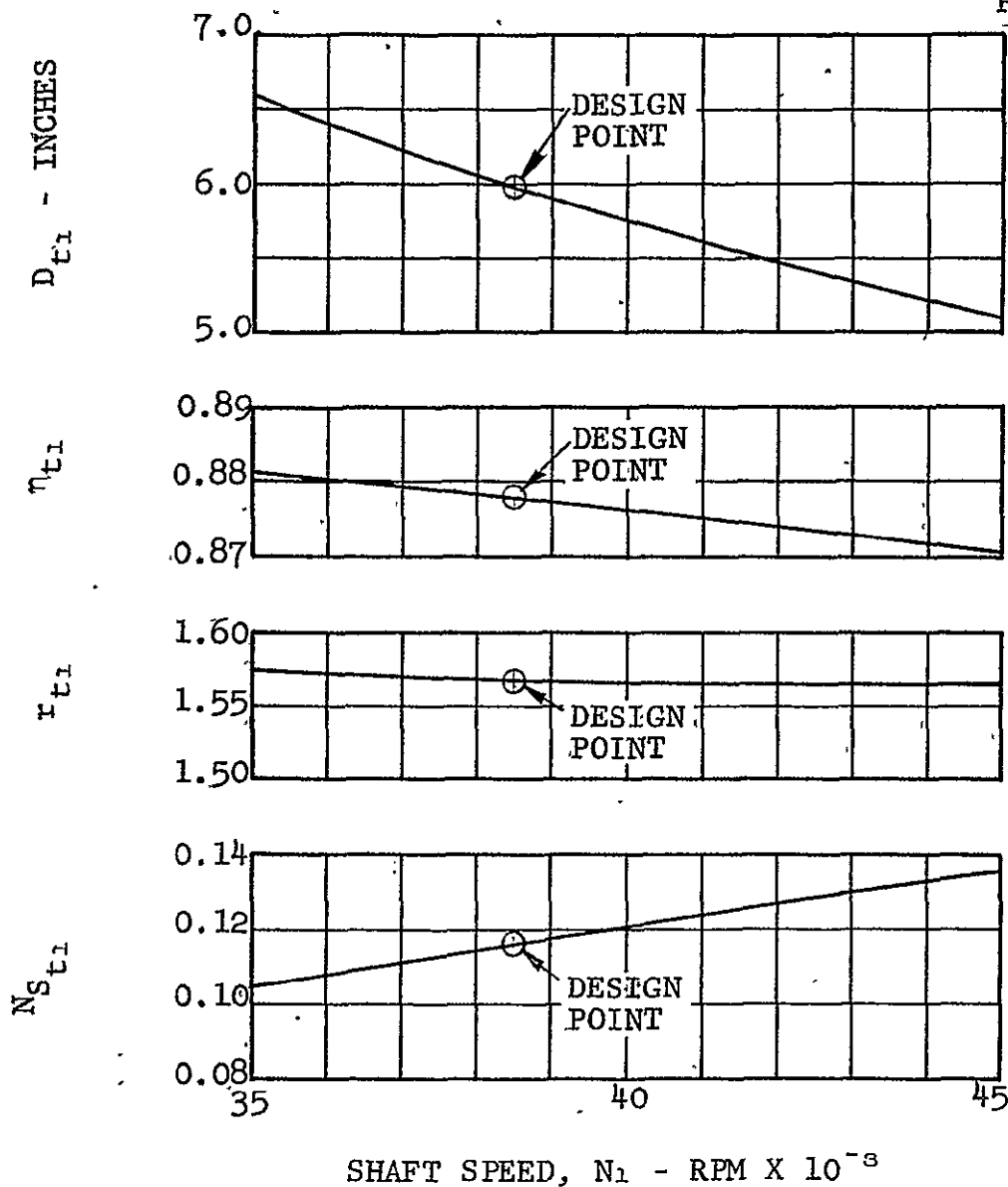
FIGURE 2.



AIRESRESEARCH MANUFACTURING COMPANY OF ARIZONA
A DIVISION OF THE GARRETT CORPORATION
PHOENIX ARIZONA

COMPRESSOR INLET TEMPERATURE, $T_1 = 536^{\circ}\text{R}$
TURBINE INLET TEMPERATURE, $T_3 = 1950^{\circ}\text{R}$
RECUPERATOR EFFECTIVENESS, $E_R = 0.85$
WORKING FLUID = ARGON

TURBINE PRESSURE RATIO
COMPRESSOR PRESSURE RATIO, $\beta = 0.90$
COMPRESSOR PRESSURE RATIO, $r_c = 2.30$
COMPRESSOR INLET PRESSURE, $P_1 = 6.0 \text{ PSIA}$
MASS FLOW RATE, $\dot{m} = 0.611 \text{ LBS PER SEC.}$



TURBINE DESIGN POINT

STUDY RESULTS

FIGURE 3

A31499-1



3.0 GAS GENERATOR DESIGN

3.1 Gas Generator Description

Figure 4 shows a schematic of the final gas generator configuration. The compressor and turbine wheels are mounted on a common shaft into which is incorporated a double-acting thrust bearing slider. Pivoted-pad gas-journal bearings support the shaft, one on each side of the thrust bearing next to the turbine and compressor wheels. These bearings are of the hydrodynamic type (self acting) with hydrostatic lift-off capability for starting and stopping. Between the journal bearing carriers are located the hydrostatic and hydrodynamic thrust bearing stators. The wheels, shaft, and bearings are mounted in the main frame, on which is also mounted the compressor scroll and diffuser and the turbine nozzle and scroll.

The following sections describe the design and development of the various components incorporated into the gas generator. Design details of the turbine and compressor have been minimized in the interest of brevity. However, the gas bearing development and integration of the components are included.

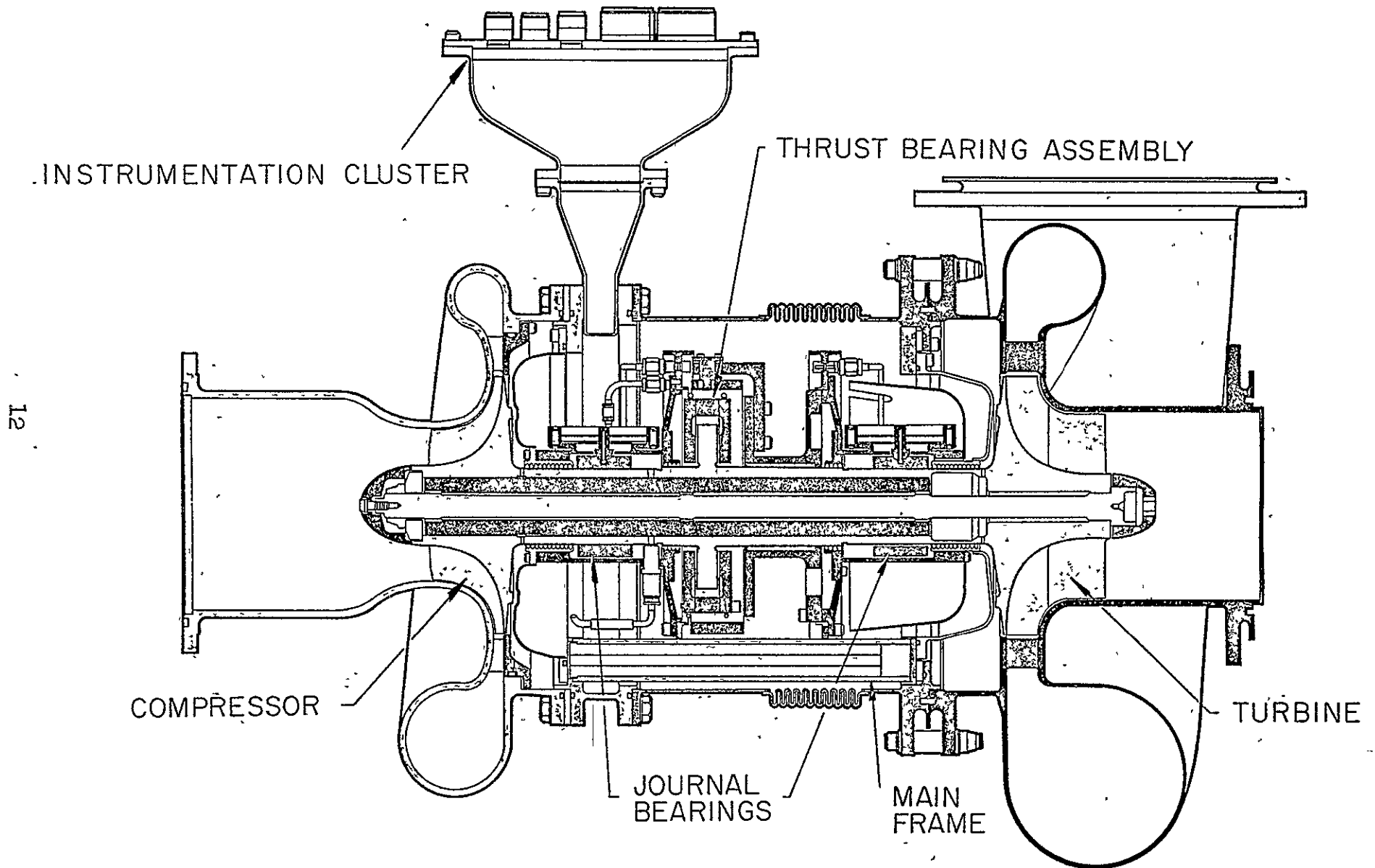
3.2 Compressor and Turbine

The compressor and turbine components of the gas generator were first designed, fabricated, and tested on cold research packages shown in Figures 5 and 6.

The compressor research package consists of a 6.0-inch-diameter radial impeller with attendant scroll and diffuser mounted on an oil-bearing test rig. Performance testing was accomplished on this rig and cutback tests were run to determine the optimum impeller cutback for the design conditions. Figure 7 shows the final map for the compressor. A complete discussion of the compressor design,



AIRSEARCH MANUFACTURING COMPANY OF ARIZONA
A DIVISION OF THE GARRETT CORPORATION
PHOENIX, ARIZONA

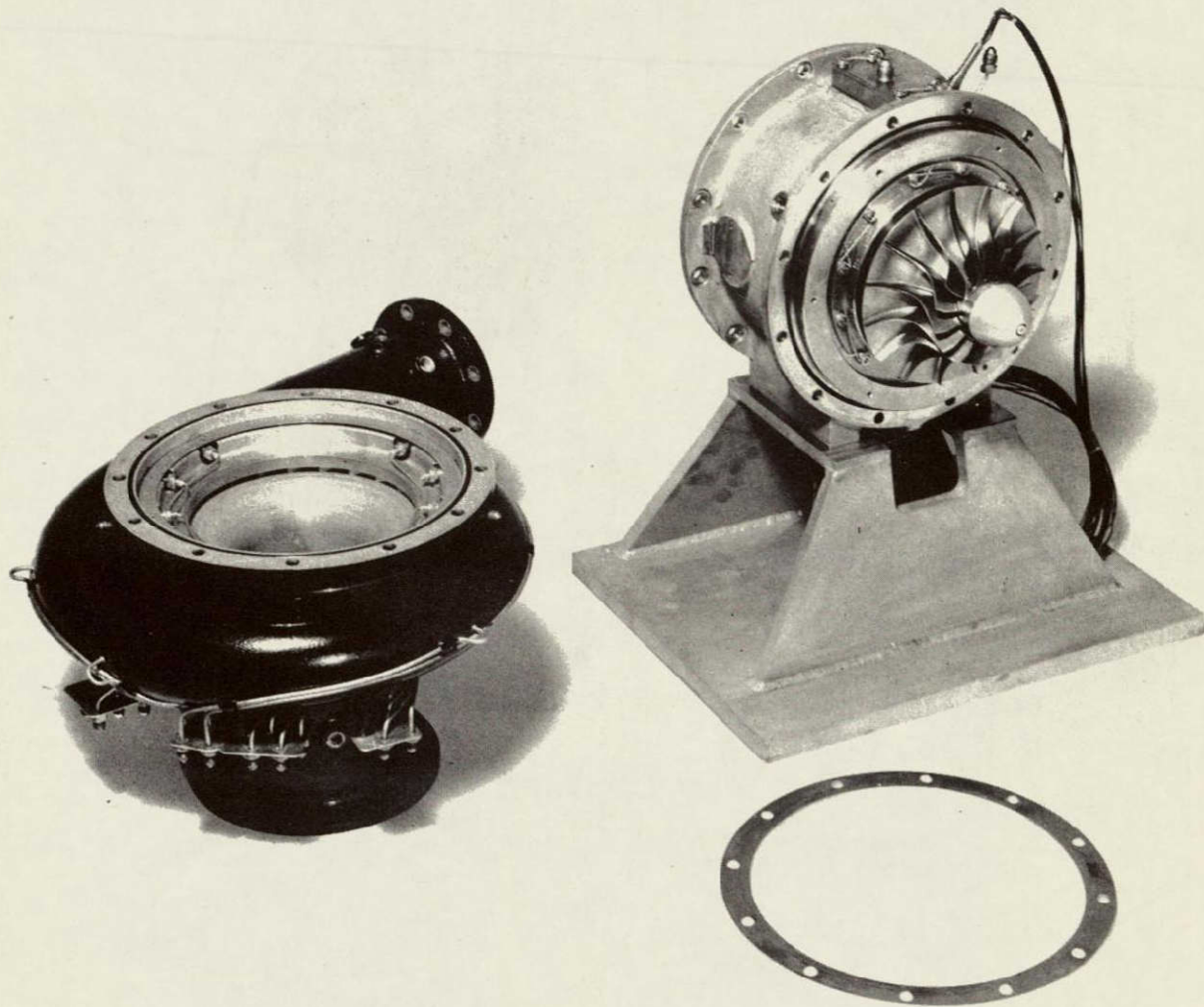


NASA BRAYTON-CYCLE RADIAL GAS GENERATOR

FIGURE 4



AIRESEARCH MANUFACTURING COMPANY OF ARIZONA
A DIVISION OF THE GARRETT CORPORATION
PHOENIX, ARIZONA

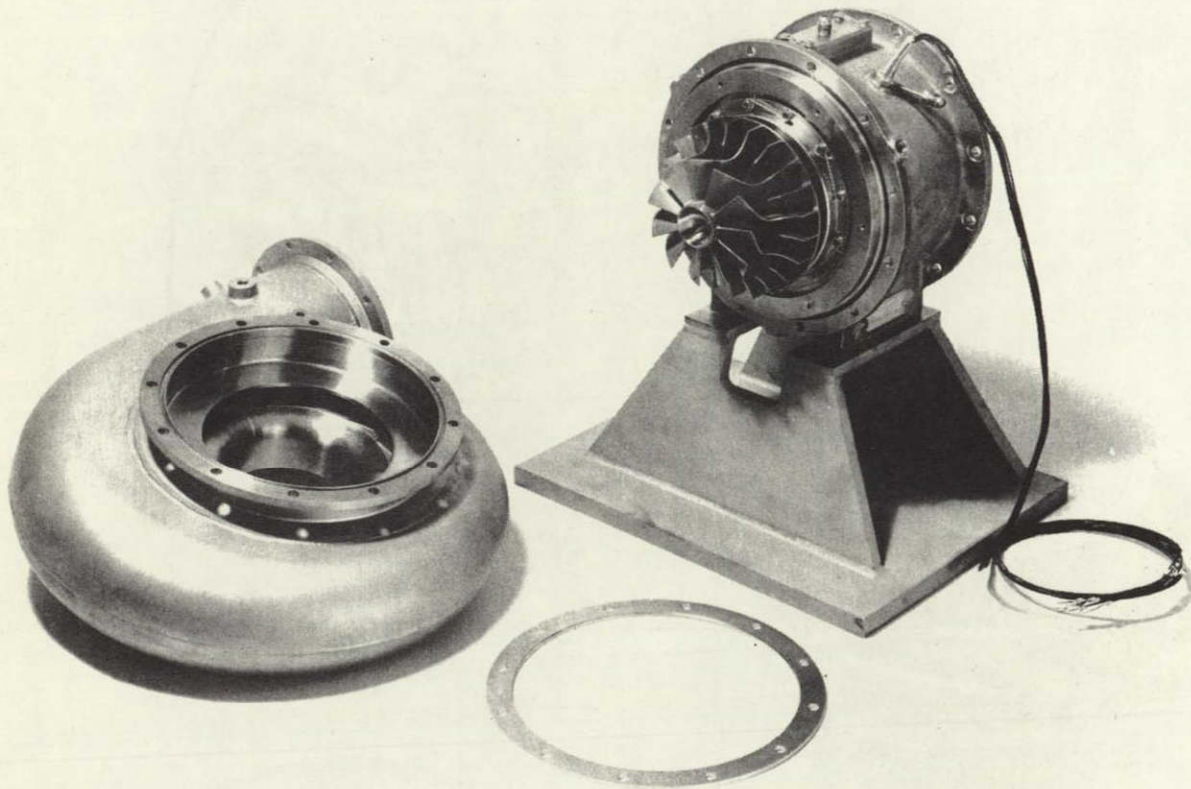


NASA COMPRESSOR RESEARCH PACKAGE

FIGURE 5



AIRESEARCH MANUFACTURING COMPANY OF ARIZONA
A DIVISION OF THE GARRETT CORPORATION
PHOENIX, ARIZONA



NASA TURBINE RESEARCH PACKAGE

FIGURE 6



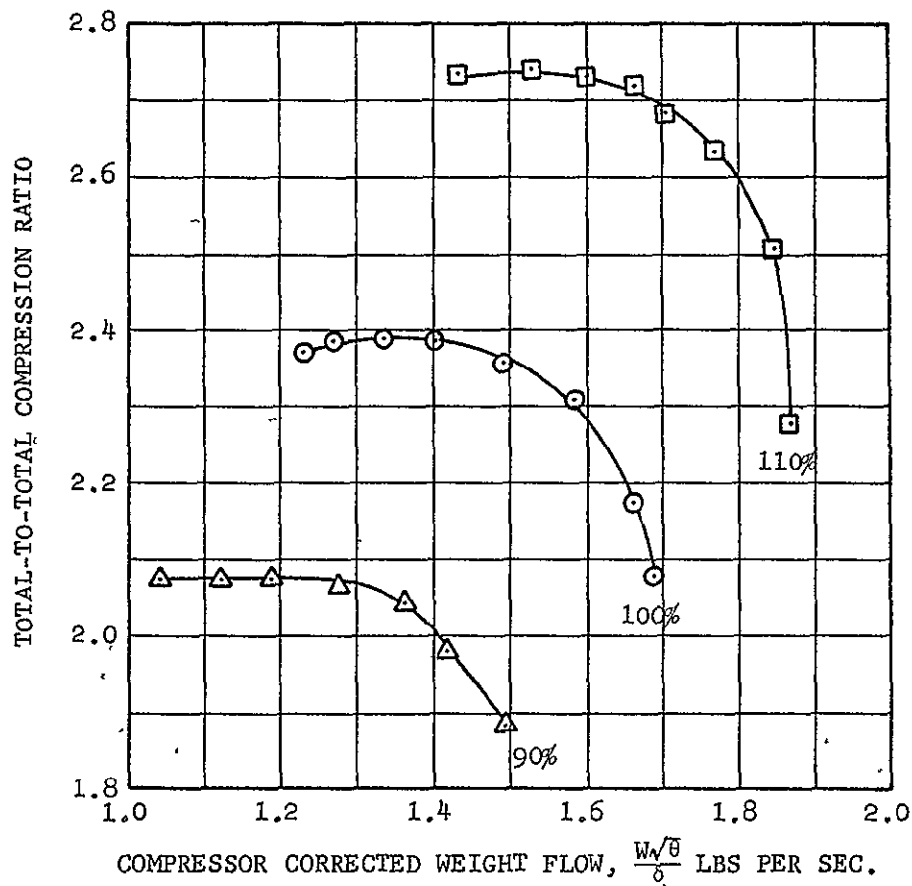
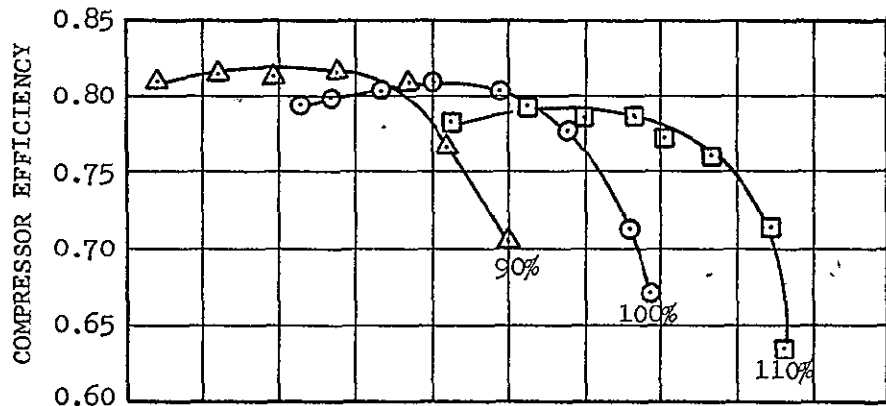
AIRESEARCH MANUFACTURING COMPANY OF ARIZONA
A DIVISION OF THE GARRETT CORPORATION
PHOENIX, ARIZONA

TEST FLUID - ARGON

COMPRESSOR INLET TOTAL PRESSURE - 12.0 TO 12.5 IN. Hg ABS.

COMPRESSOR INLET TOTAL TEMPERATURE - 536°R

100 PERCENT $N/\sqrt{\theta} = 37,900$ RPM



6-IN.-DIAMETER COMPRESSOR TEST RESULTS
FIGURE 7

A31391-1



fabrication, and acceptance testing can be found in NASA Report CR-54368 entitled Design and Development of a High-Performance Brayton-Cycle Compressor Research Package (AiResearch Report APS-5109-R). Two compressor research packages were delivered to the NASA.

The turbine research package consists of a 6.0 inch diameter radial turbine wheel with attendant scroll and nozzle mounted on an oil-bearing test rig. Development testing was accomplished on this rig. A complete discussion of the turbine design, fabrication, and acceptance testing can be found in NASA Report CR-54367 entitled Design and Development of a High-Performance Brayton-Cycle Turbine Research Package (AiResearch Report APS-5108-R). Two turbine research packages were delivered to the NASA.

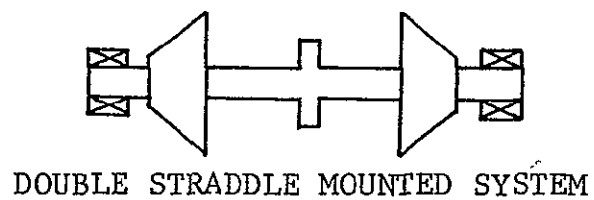
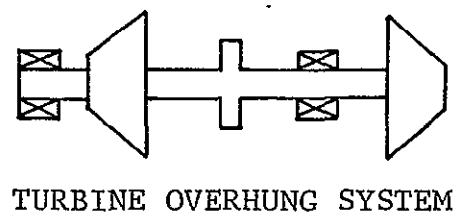
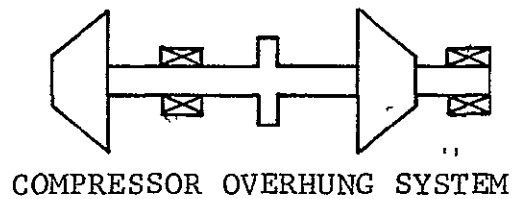
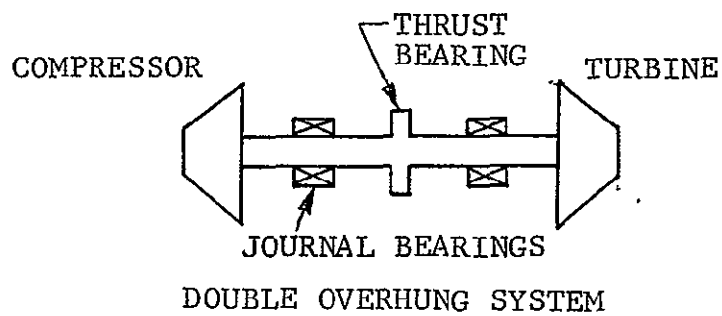
3.3 Dynamic Analysis

At the initiation of the gas-generator design, four possible gas-bearing/turbine and compressor wheel configurations were analyzed to determine the best configuration from critical speed and design considerations. The four systems, shown in Figure 8, consisted of a double overhung system, a compressor overhung system, a turbine overhung system, and a system wherein the bearings are mounted outside the turbine and compressor wheels. Of the four configurations analyzed, the double overhung system was chosen for continued analysis since it presented fewer overall design problems.

A bearing load and critical speed analysis was then conducted for the double overhung system with a range of bearing spacings (6.0 to 14.5 inches) and overall resilient gas bearing spring rates (5,000 to 20,000 pounds per inch). The resiliently mounted



AIRESEARCH MANUFACTURING COMPANY OF ARIZONA
A DIVISION OF THE GARRETT CORPORATION
PHOENIX, ARIZONA



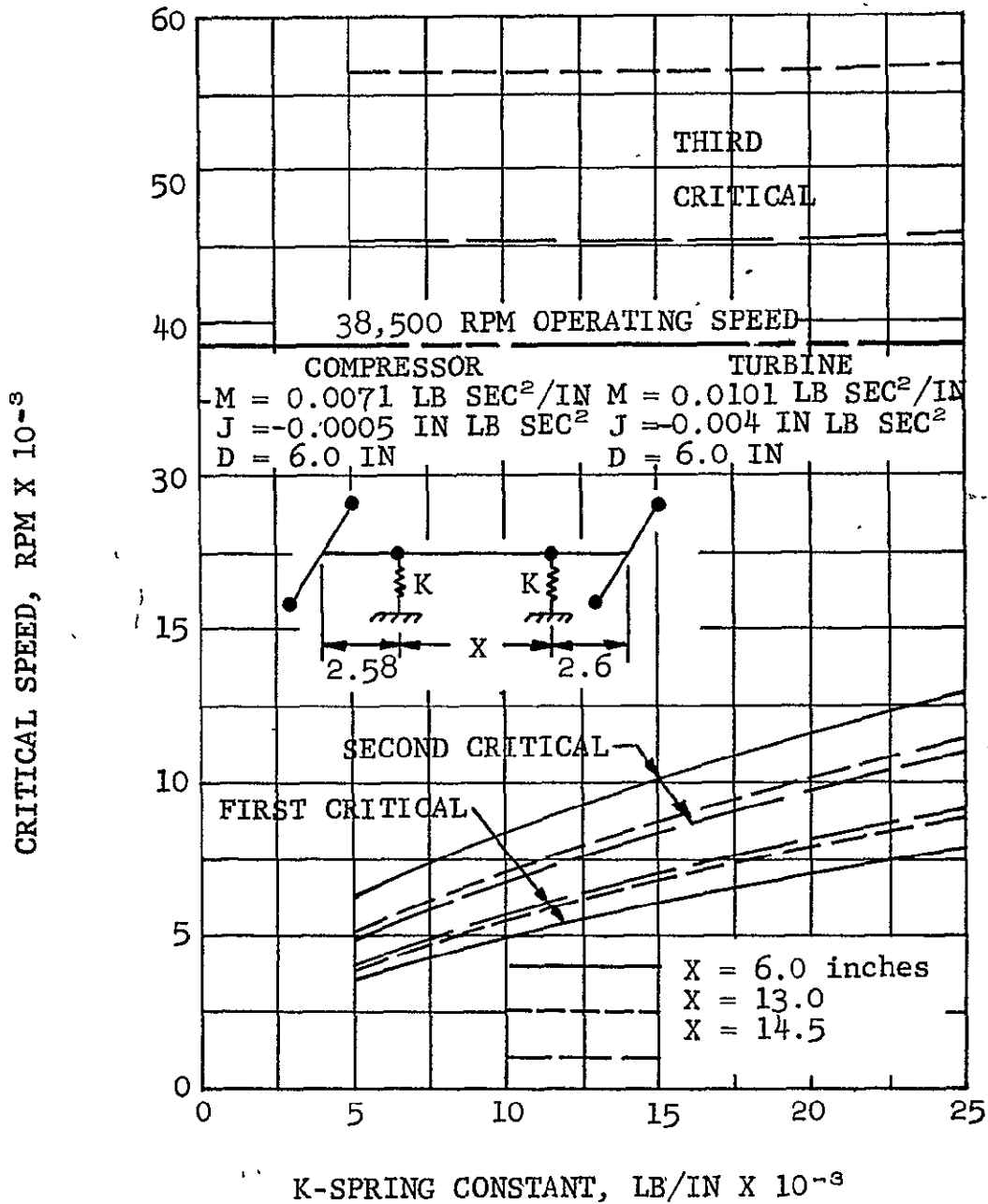
GAS GENERATOR CONFIGURATIONS
FIGURE 8



bearings tend to minimize the critical speeds and bearing loads (see Section 3.4 for a discussion of the resiliently mounted bearings). Figure 9 shows the results of the critical speed analysis. For the variable spacing and bearing combined gas film and mount spring rates, the bearing loads at design operating speed were approximately 20 pounds per 0.001 inch c.g. eccentricity (i.e., the c.g. of both the turbine and compressor masses are displaced 0.001 inch in the same direction from the rotational center). Bearing spacings from 6.0 to 13.0 inches were initially chosen to be utilized in the gas bearing testing and as limits for the gas generator. The 14.5-inch bearing spacing was rejected since the third critical speed is too close to the 38,500 rpm operating speed.

These results were obtained by utilizing an AiResearch critical-speed computer program. This program has the capability of considering gyroscopic moments, shaft flexibilities, bearing (and support) flexibilities, static and dynamic rotor unbalance, and rotor distributed properties. The rotor may be lumped into a maximum of 30 mass stations, and at each mass station the mass, polar moment of inertia, diametral moment of inertia, mass unbalance and bearing flexibility may be specified. The whirl ratio may also be prescribed. The program computes bearing loads versus speed and the rotor critical speeds and corresponding rotor modal patterns.

After the design of the gas generator was finalized, the final critical-speed analysis was accomplished, and the results are shown in Figure 10. For this analysis the bearing spacing was 7.0 inches, and the combined rotating group polar moment of inertia was 0.060 inch-lbs-sec².



CRITICAL SPEED VS. BEARING SPRING RATE FOR
NASA GAS GENERATOR, INITIAL RESULTS

A30726-1

FIGURE 9

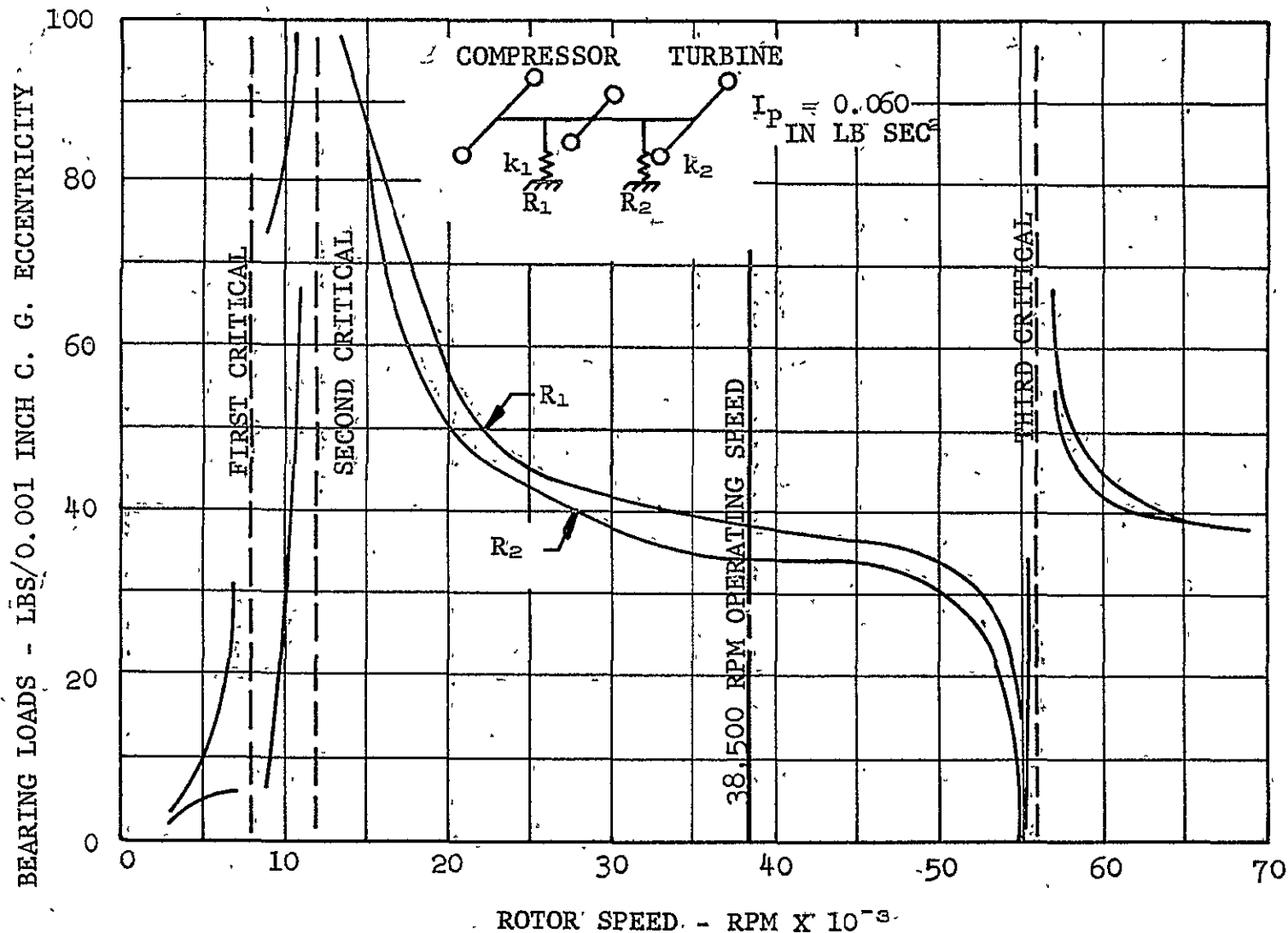


AIRESEARCH MANUFACTURING COMPANY OF ARIZONA
A DIVISION OF THE GARRETT CORPORATION
PHOENIX, ARIZONA

$e_1 = e_2 = 0.001$ INCH

$X =$ BEARING SPACING $= 7.0$ INCHES

$k_1 = k_2 = 35,800$ POUNDS PER INCH (GAS FILM STIFFNESS)



BEARING LOAD VS. ROTOR SPEED
NASA GAS GENERATOR FINAL CONFIGURATION

A32572

GROUP 0



AIRESEARCH MANUFACTURING COMPANY OF ARIZONA
A DIVISION OF THE GARRETT CORPORATION
PHOENIX ARIZONA

It should be noted that the bearing loads shown in Figure 10 are for c.g. eccentricities of 0.001-inch. However, the c.g. eccentricity obtained on both the bearing test rig shafts and the gas generator is in the range of 0.0001-inch which reduces the actual bearing loads by a factor of 10.



3.4 Journal Gas Bearings

3.4.1 Design Considerations and Bearing Selection

The use of gas-lubricated bearings for the rotor support system of the gas generator is contingent upon three requirements:

- (a) that an effective means be provided to accommodate dimensional changes and misalignments of the components associated with the bearings.
- (b) that rotor dynamic stability be ensured throughout the system operating range.
- (c) that the bearings can be self-acting (use process gas) at the rotor design speed (no external pressurization available for long-time operation).

A careful review of the above requirements led to the selection of self-acting partial-arc pivoted-pad bearings. This type of bearing offers attractive mechanical design features such as the ability to accommodate rotor radial misalignments and minor bearing surface imperfections. In addition, rotor dynamic stability is, in general, considerably improved with pivoted shoe bearings as compared to other bearing types such as conventional cylindrical journal bearings.

Based on extensive turbomachinery experience with both anti-friction bearings and conventional oil journal bearings, AiResearch chose to resiliently support the two journal bearing assemblies of the gas generator. The following desirable advantages are realized in the use of resilient mounts:



- (a) Rigid rotor critical speeds are influenced by resilient bearing mounts. Proper choice of the support spring rate will ensure that the first two system critical speeds are in the low-speed, low-energy regime.
- (b) Large transient rotor dynamic motions, such as occur during acceleration through system critical speeds, are not limited by the gas film thickness, since the resilient mounts will accommodate large movements.
- (c) Rotor and bearing dimensional variations as a result of thermal and centrifugal growth are partially accommodated by the deflection of the resilient mounts.
- (d) The ability to establish a positive preload on the rotor-bearing group to suppress whirl tendencies in "zero-g" environments (space power systems).

The last advantage cited--that of establishing a positive bearing preload (shoe-rotor interference at zero speed), brings about an additional design consideration of its own--that of external pressurization (a hydrostatic shoe lift-off flotation system) for starting and stopping. This combination of external pressurization during starting and stopping and self-acting ability during normal operation (no external pressurization required) eliminates the possibility of any contact between the tilting shoes of the journal bearing and the surface of the rotor.

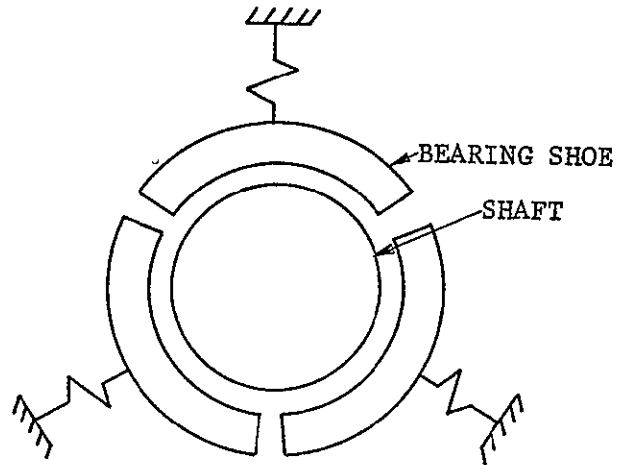
The bearing configuration selected for the gas generator bearing development consisted of three bearing shoes, with each shoe encompassing a 100-degree arc. All the shoes were provided with pivotal freedom, and one or more of the shoes at each bearing were provided with radial resiliency. The shoes initially contacted



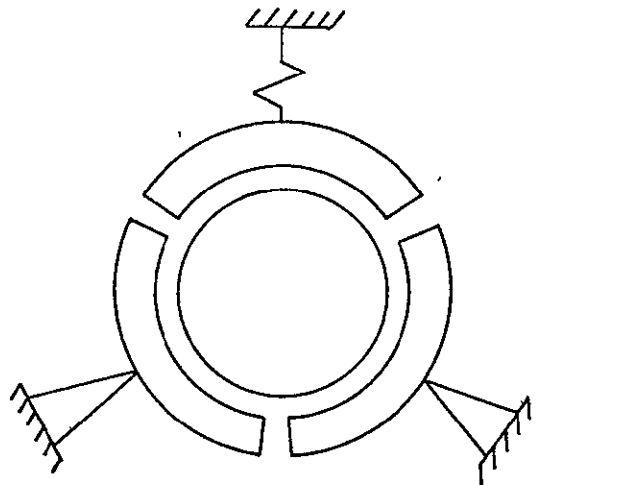
the journal with a predetermined interference load (preload) which was accommodated by the shoe resilient supports. Each shoe was lifted from the journal prior to rotation by means of hydrostatic orifices incorporated in the bearing shoe surfaces. The hydrostatic gas supply was then removed from the orifices when the shaft assembly had reached its design operating speed.

Figure 11 schematically illustrates the two resilient mounting methods that were developed. In Case 1, each of the shoes is resiliently supported, and the differential dimensional changes are accommodated by the flexibility of each of the lubricant films and of each of the three shoe supports. The equivalent mechanical system simulating this case is one of each shoe being separated from the journal by a nonlinear spring in a parallel with a nonlinear viscous damper. In this case the total film resistance acts in series with the shoe resilient support linear spring. In Case 2 of Figure 11, only one shoe at each bearing is resiliently supported, and all of the differential dimensional change is accommodated within each lubricant film and the single-shoe resilient support.

Figure 12 shows two basic bearing shoe and mount configurations that were evaluated during the testing program. Both the fixed-stem (elastically pivoted) and the ball-and-socket resiliently mounted gas bearing designs utilize the deflection of the circular plate or diaphragm to provide the mount spring rate. The inherent angular stiffness of the fixed-stem design is eliminated by the ball-and-socket design. The hydrostatic lift-off system consists of either a single orifice or four orifices incorporated into the bearing shoe surface.

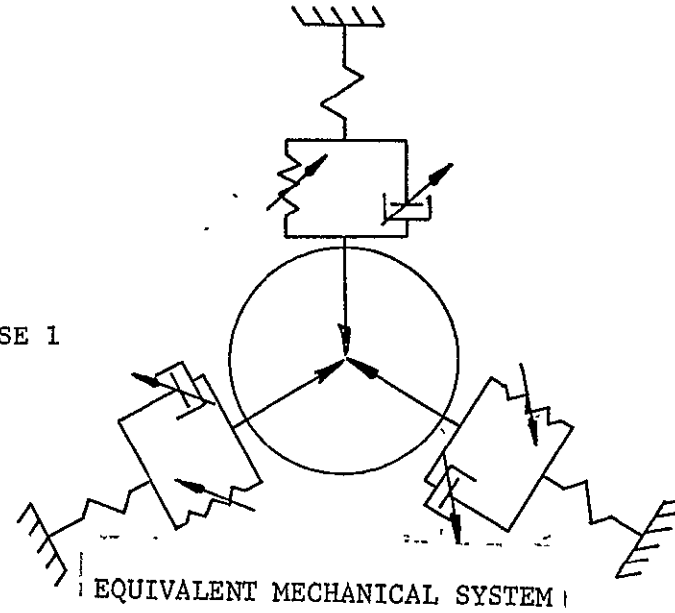


THREE SHOES FLEXIBLY MOUNTED



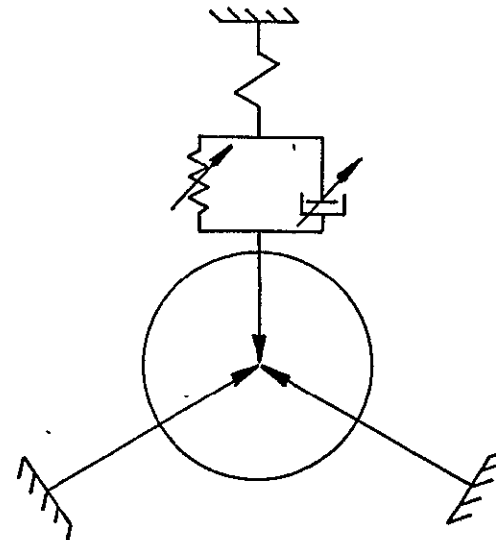
TWO FIXED MOUNTS, ONE FLEXIBLY MOUNTED

CASE 1



EQUIVALENT MECHANICAL SYSTEM

CASE 2



EQUIVALENT MECHANICAL SYSTEM

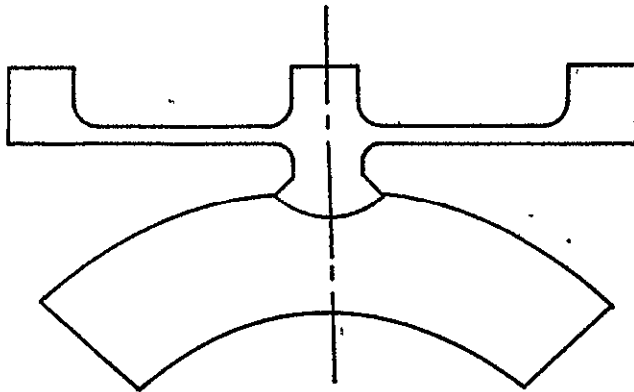
BEARING MOUNTING METHODS
FIGURE 11

A31813

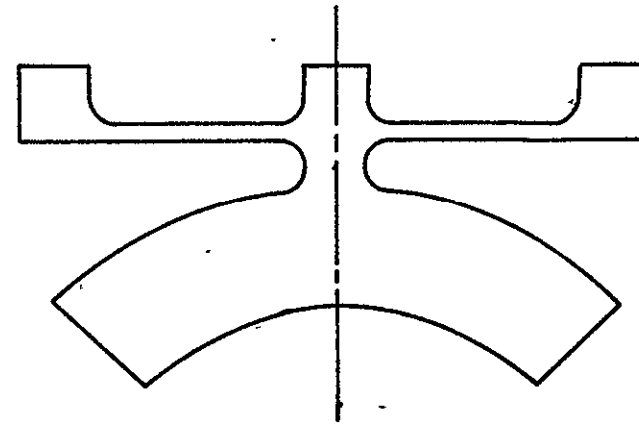


AIRESEARCH MANUFACTURING COMPANY OF ARIZONA
A DIVISION OF THE GARRETT CORPORATION
PHOENIX, ARIZONA

26



BALL-AND-SOCKET
DIAPHRAGM MOUNT



FIXED-STEM
DIAPHRAGM MOUNT

GAS BEARING SHOE MOUNTING METHODS

FIGURE 12

A31809



3.4.2 Gas Bearing Analysis

Once the aerodynamic performance of the gas generator was determined (compressor impeller and turbine wheel sizes) preliminary layouts of the main rotating assembly were completed. Tentative bearing span ranges from 6.0 to 13.0 inches were obtained from these layouts. The mass and inertial properties of the aerodynamic components, and assumed rotor dimensions, including the range of bearing spans, plus a range of bearing support flexibilities, were used to determine a range of rigid-body critical speeds, as well as the third critical speed (free-free bending mode).

In general, the rotor diameter size is the governing criterion to keep the shaft free-free bending mode (third critical speed) well above the operating speed. The gas bearing journal size was therefore determined from the above analysis. The final task was to determine whether gas bearings of the prescribed diameter had adequate self-acting load-carrying ability, gas film stiffness, etc., to meet the bearing load requirements.

Analysis of the gas bearings was accomplished by a steady-state digital computer program that is capable of considering either incompressible or compressible fluids operating in either the laminar or turbulent flow regimes. Either a full journal or a single sector may be prescribed with local variation in the surface geometry. In addition, provision is made for considering local hydrostatic lift-off orifices. Thus, externally pressurized, self-acting, or hybrid lubrication may be analyzed for a single shoe.



An iterative numerical technique is employed to solve the Reynolds equation for finite length bearings, and, thus, end-leakage effects are implicit in the analysis. Any combination of journal dynamic motions may be considered that would introduce misalignment, journal eccentricity rate of change, and journal whirl. The program predicts bearing loads, center of pressure coordinates, and moments about any specified set of coordinate axes, flow-leakage rates, and pressure distribution for a maximum matrix size of 21 rows and 36 columns.

The journal bearing computer program was utilized to formulate the initial bearing designs committed to the test program. In addition, the computer program greatly simplified the problem of evaluation of test results.

The test rotors were evolved from the aforementioned analysis. The bearing spans ranged from 6.0 to 13.0 inches. The rotors were hollow with approximately 1/4-inch wall thickness at the journals. The journal surfaces were Flame-Plated with tungsten carbide. The mass and inertial properties of each were similar to those of the future gas generator rotating assembly. For the 11-inch bearing span rotor (most frequently used in development testing), these properties were:

Mass: $0.0384 \text{ lb sec}^2 \text{ per inch}$

Polar moment of inertia: $0.033 \text{ in.-lb-sec}^2$

Diametral moment of inertia: $1.271 \text{ in.-lb-sec}^2$



The pivoted-pad gas bearing/resilient-mount configurations initially analyzed and tested were:

Pivot = ball/socket and fixed-stem (elastic pivot)

Mount stiffnesses = 20,000 lbs per inch and 10,000 lbs per inch

Lift-off system = 4-orifice and single-orifice

Clearance ratio (C/R) = 0.001

Journal radius (R) = 1.000 inch

Angular span (degrees) = 100

L/R ratio = 1.5

Pivot location = 50° from leading edge

3.4.3 Gas Bearing Testing

Development testing of the gas bearings was accomplished on two development test rigs--the single-shoe test rig, and the dynamic two-bearing test rig. The single shoe test rig was used to investigate isolated behavior of the bearing shoe, such as hydrostatic capability, hydrodynamic performance, misalignment effects, pivotal function, shoe instability, etc., of some of the bearing designs considered in Section 3.4.3.3. The two-bearing test rig was used to evaluate complete bearing systems with the shaft oriented in the vertical plane, since this best simulated zero-g conditions for journal bearings and represented the most critical bearing operation from the standpoint of stability. All testing was undertaken in ambient air conditions, and hydrostatic supplies utilized pressurized air which could be varied from zero to 160 psig as necessary.

3.4.3.1 Single-Shoe Test Rig

This test rig consists of a precision surface rotor mounted on precision ball bearings and driven by an air turbine. The rotor has two capacitance probes imbedded in the surface 180 degrees apart and spaced to sweep under the outer edge of each of the test shoes. Both probes are commonly connected to an axially located



junction which terminates at one end of the rotor. A spring-loaded needle brush, bearing against the rotor end, conveys the capacitance signal to a remotely located distance meter (Photocon Products Dynagage). Thus each revolution of the rotor produces two signals (sweep of each shoe edge) and gives a continuous display of the clearance between the shoe and the rotor. The test bearing shoe is mounted in a bearing carrier and loaded against the rotor. Strain gauges, mounted in a bridge circuit on the bearing resilient circular plate, are used to establish the desired initial preload, and for constantly monitoring bearing operating loads. Figure 13 shows the single shoe test rig.

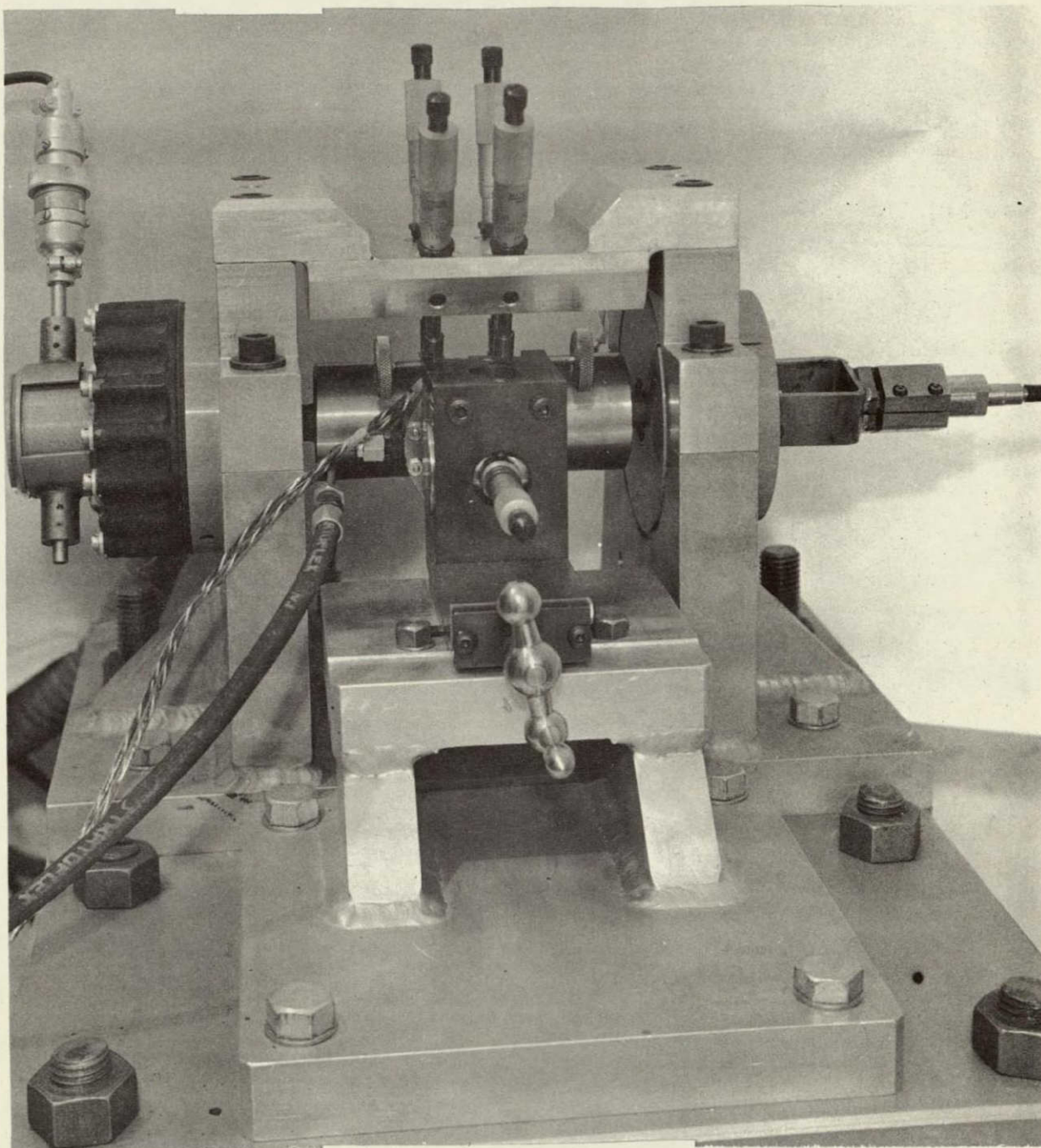
3.4.3.2 Dynamic Two-Bearing Test Rig

The two-bearing rotor dynamic test rig, shown in Figure 14, is used to investigate the dynamic behavior as well as other aspects of the system comprising the gas bearings and test rotor.

The test rotor is vertically supported in the test rig and is driven by a turbine wheel mounted at the rotor upper end. A hydrostatic thrust bearing supports the rotor at the bottom. Each bearing shoe assembly is secured to a support assembly. This support assembly is mounted to a track directed radially with respect to the rotor, and radial positioning of the assembly is provided by a calibrated screw thread. The track may also be pivoted about an axis parallel to the rotor axis. In addition, the support assembly has provision for pivoting of the bearing shoe assembly about a horizontal axis directed perpendicularly to the track. Thus, the angular attitude of each bearing shoe assembly may be adjusted mechanically about three orthogonal axes, and any desired initial clearance or interference between shoe and shaft may be established. A micrometer adjustment in each support assembly limits the maximum radial deflection of the bearing resilient support. Any type of resilient supporting system with any desired flexibility may be tested with this rig.



AIRESEARCH MANUFACTURING COMPANY OF ARIZONA
A DIVISION OF THE GARRETT CORPORATION
PHOENIX, ARIZONA



SINGLE-SHOE BEARING TEST RIG

FIGURE 13

**AIRESEARCH MANUFACTURING COMPANY**

A DIVISION OF THE GARRETT CORPORATION

PHOENIX, ARIZONA

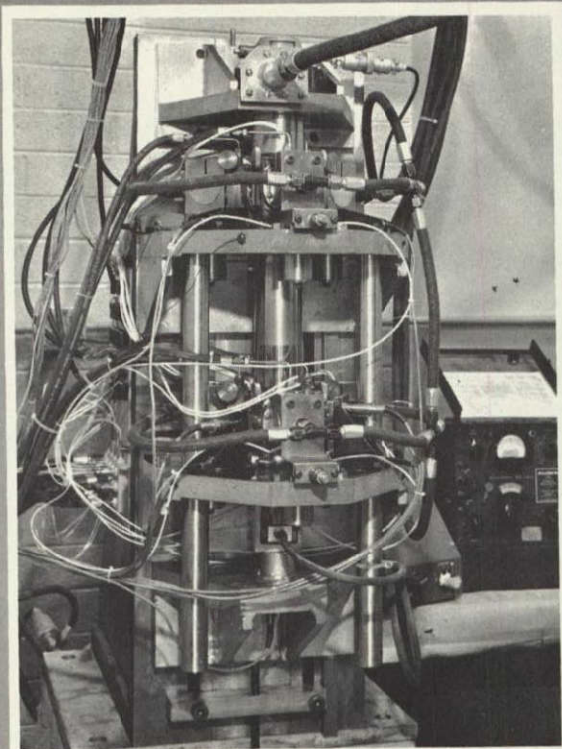
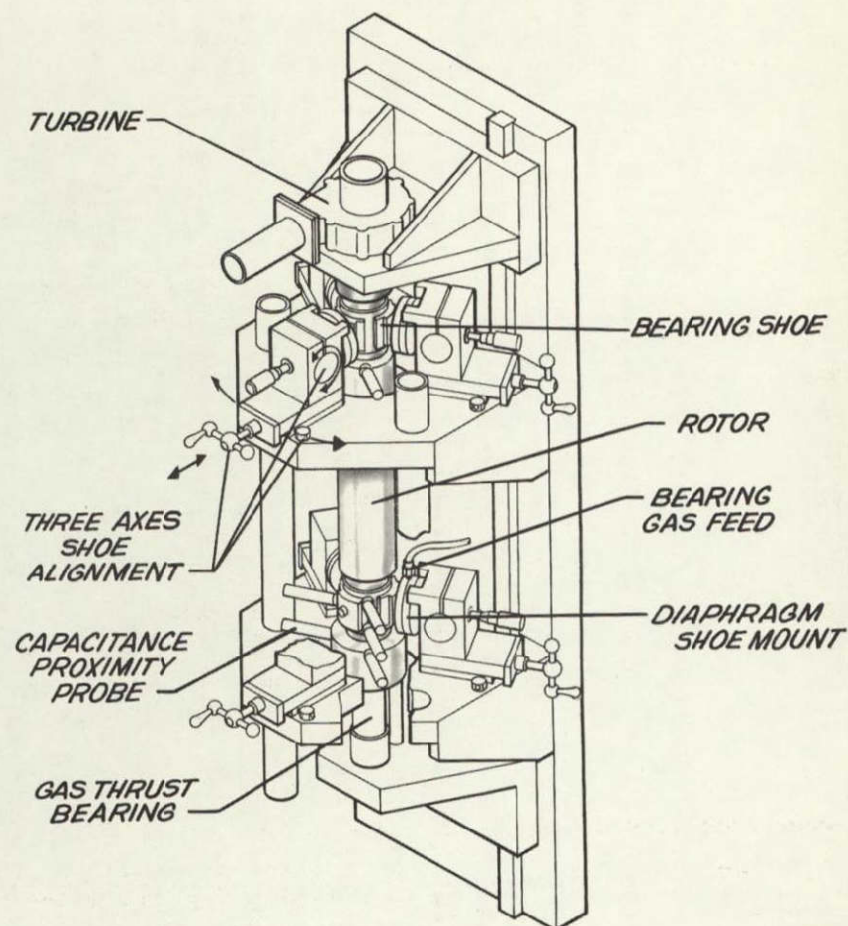


FIGURE 14

32



DYNAMIC TWO-BEARING TEST RIG



The test-rig bearing support brackets are made so that the bearing span may be varied to accommodate rotors of various lengths.

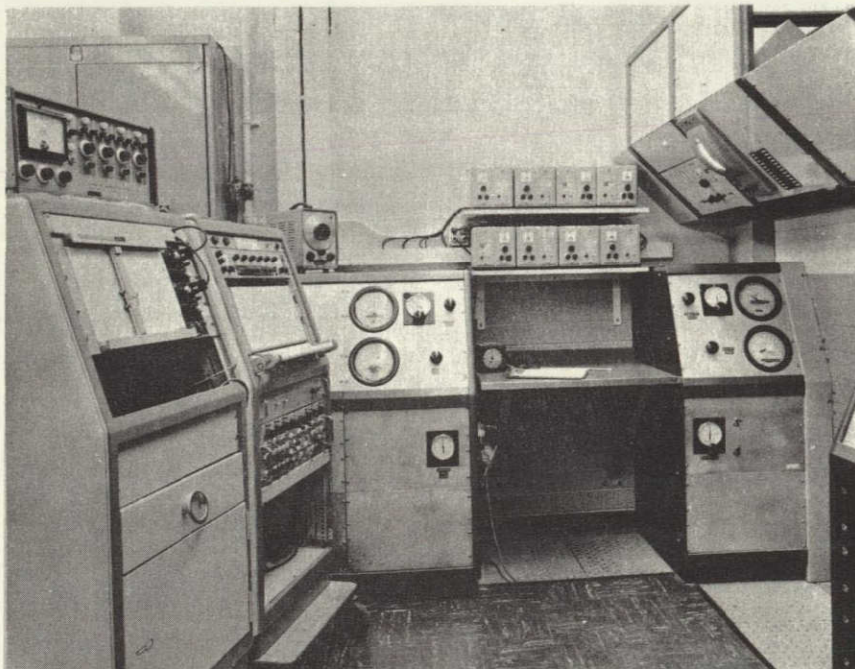
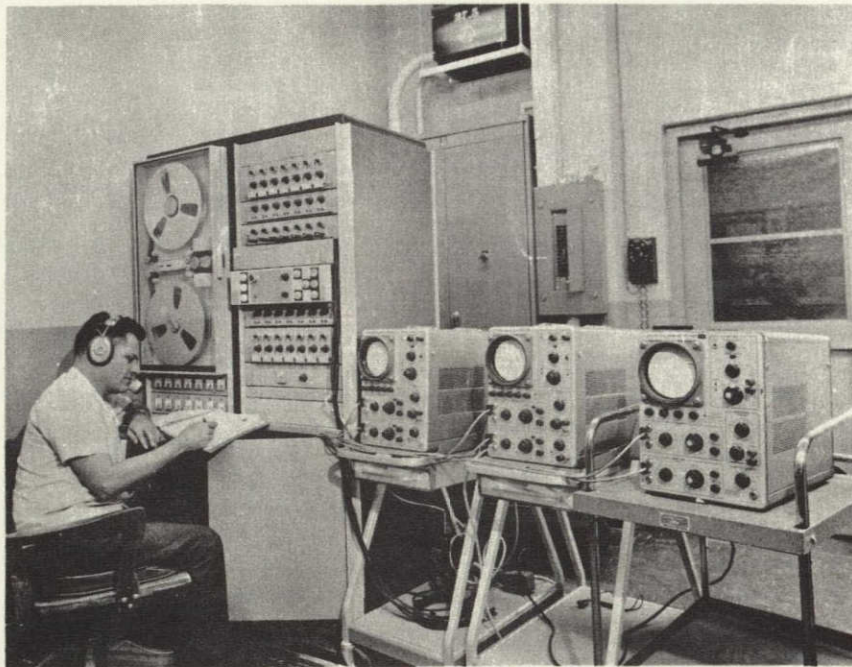
Capacitance probes (Photocons), mounted in quadrature with respect to the rotor axis on each bearing bracket, monitor shaft dynamic motions in the radial direction near each bearing. In addition, the relative motions of one bearing shoe are monitored by two capacitance probes directed at the upper and lower regions of the shoe leading edge, and also by a third Photocon directed at the shoe trailing edge, thus providing the capability of describing the motion of the shoe in the two planes of pitch and roll. A Photocon at the bottom of the rotor measures the axial motions of the rotor. This instrumentation, as well as speed output, is connected to a 14-channel magnetic tape recorder (see Figure 15 for the bearing test facility instrumentation), thus providing a means of recording the continuous history of test runs from start-up to full speed.

For each test, initial alignment of each bearing shoe may be obtained with the test rotor supported on alignment center pins at each end. After alignment is achieved, the center pins are removed, and the hydrostatic thrust bearing is replaced at the bottom of the shaft. Strain gauges cemented to each bearing shoe resilient plate are calibrated as functions of both linear and angular displacements of the bearing. The outputs from each of these strain gauges are utilized as the basis of establishing the desired radial alignment and initial preload between the bearing and the shaft.

These strain-gauge signals are also utilized to continuously monitor bearing loads during testing. Thermocouples connected to the bearing shoes are used to determine nominal shoe temperature, and signals from the thermocouples and strain gauges are continuously recorded on a Sanborn recorder.



AIRESEARCH MANUFACTURING COMPANY OF ARIZONA
A DIVISION OF THE GARRETT CORPORATION
PHOENIX, ARIZONA



GAS-BEARING TEST FACILITY INSTRUMENTATION
FIGURE 15



3.4.3.3 Test Results

As previously noted on page 29, the four basic pivoted-pad gas bearing configurations initially tested were (in order):

- (1) Fixed-stem, four-orifice
- (2) Ball-and-socket, four-orifice
- (3) Fixed-stem, single-orifice
- (4) Ball-and-socket, single-orifice

Each of the above bearing configurations utilized the equal spring mount system shown in Figure 11, Case 1.

The tests of the first bearing configuration (fixed stem, four orifice) were satisfactory and rotor speeds to 50,000 rpm were accomplished without incident. These tests established that a bearing of this type might meet the operating requirements. However, initial alignment problems inherent in the fixed-stem design led to further testing, the goal being to develop a workable ball-and-socket pivot design to alleviate the bearing alignment problems in the final gas generator. Test evaluation of the remaining three configurations indicated that a shoe geometry change was required to provide a wide range of stable rotor operation in the externally pressurized, hybrid, and self-acting bearing regimes. Some of the problems encountered in these tests were shoe flutter and spragging (shoe leading-edge dip).

Results of computer analyses showed that an increase in the shoe clearance ratio and a relocation of the shoe pivot point would tend to optimize the shoe stability characteristics and also provide a reduction in the predicted shoe friction loss.



Based on this information, the bearing shoes were redesigned (both fixed-stem and ball-and-socket types) with the pivot located 65 percent back from the shoe leading edge and the clearance ratio (C/R) increased to 0.0017. Testing of these configurations with the equal spring mount system proved to be satisfactory. The shoe flutter and spragging mentioned above were no longer a problem. The spring rate of this particular system was in the order of 20,000 pounds per inch. In order to decrease the mount spring rate, the mounting system depicted in Figure 11, Case 2, was used (two fixed shoes per bearing and one resilient mount). The composite spring rate of this system was in the order of 5,400 to 6,000 pounds per inch. The stability of this system was excellent, and in addition, the low composite spring rate afforded adequate rotor thermal and centrifugal growth relief.

3.4.4 Gas Generator Journal Bearing Configuration

As a result of the testing discussed above, the gas generator bearing configuration selected was based on a similarity of the bearing compressibility number, λ , between the gas generator bearing and the air test bearings where

$$\lambda = \frac{6\mu\Omega}{P_a} \left(\frac{R}{C}\right)^2$$

μ = lubricant viscosity, lbs per sec per sq inch

R = journal radius, inches

Ω = angular velocity, radians per second



P_a = ambient pressure, psia

c = bearing clearance (difference between bearing and journal radii), inches

λ = bearing number

For the gas generator with a design lambda (λ), ≈ 1.5 , the journal bearing configuration was thus determined to be:

Journal radius, inches = 0.875

Clearance ratio = 0.00263

L/R ratio = 1.50

Angular span, degrees = 100

Pivot location (from leading edge), degrees = 65

Pivot type: Ball-and-socket

Mounting system: Two shoes per bearing radially constrained, the third shoe resiliently supported by a 2,500-pound-per-inch diaphragm mount

Number of hydrostatic supply orifices per shoe: Four



These characteristic requirements are based on the machine design point of 38,500 rpm shaft speed, 12 psia journal ambient pressure, argon gas atmosphere, and a journal ambient temperature level of 500°F.

The four hydrostatic orifices provide easier lift-off action and are utilized instead of the single orifice so that the orifices are located away from the point of peak hydrodynamic pressure.

Figure 16 shows the predicted performance of the bearings at the design conditions.

3.5 Thrust Gas Bearings

3.5.1 Thrust Bearing Design

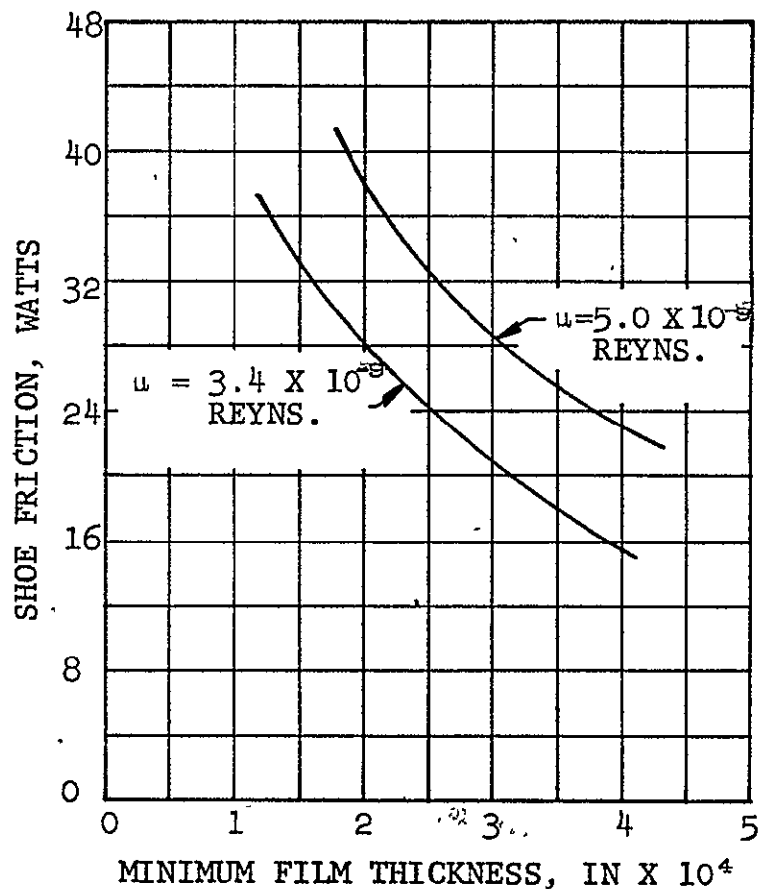
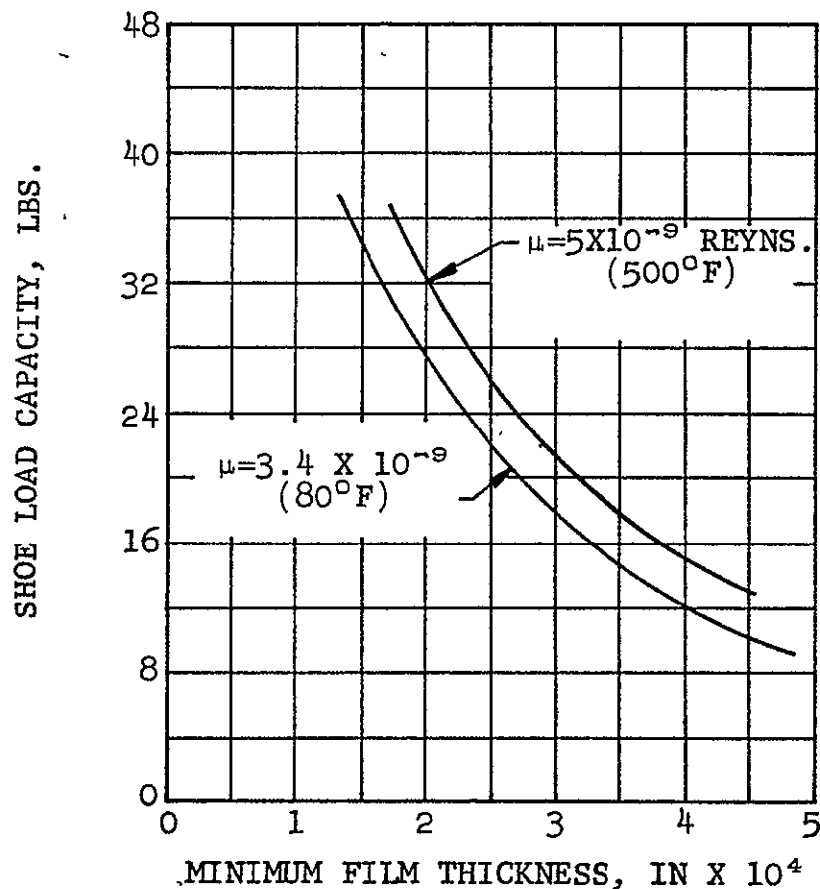
The design of the thrust bearing was based on the following calculated gas generator operating requirements when operating in the design system:

- (a) A maximum start-up thrust load to self-sustained speed of 60 pounds in a direction from the compressor to the turbine (with a turbine back pressure of 3 psia assumed).
- (b) At self-sustained speed (approximately 21,000 rpm), the thrust direction reverses and its magnitude becomes approximately 20 pounds.
- (c) The thrust load increases from 20 pounds to a maximum value of 30 pounds at operating speed, in a direction from the turbine to the compressor. The normal thrust bearing is to operate hydrodynamically, beginning at self-sustained speed.



BEARING CONFIGURATION

1. ANGULAR SPAN - 100 DEGREES
2. $L/R = 1.5$
3. AMBIENT PRESSURE = 12 PSIA
4. $R = 0.875$ INCHES
5. ARGON AT TWO TEMPERATURES



PREDICTION OF GAS GENERATOR JOURNAL BEARING
PERFORMANCE AT DESIGN SPEED

FIGURE 16



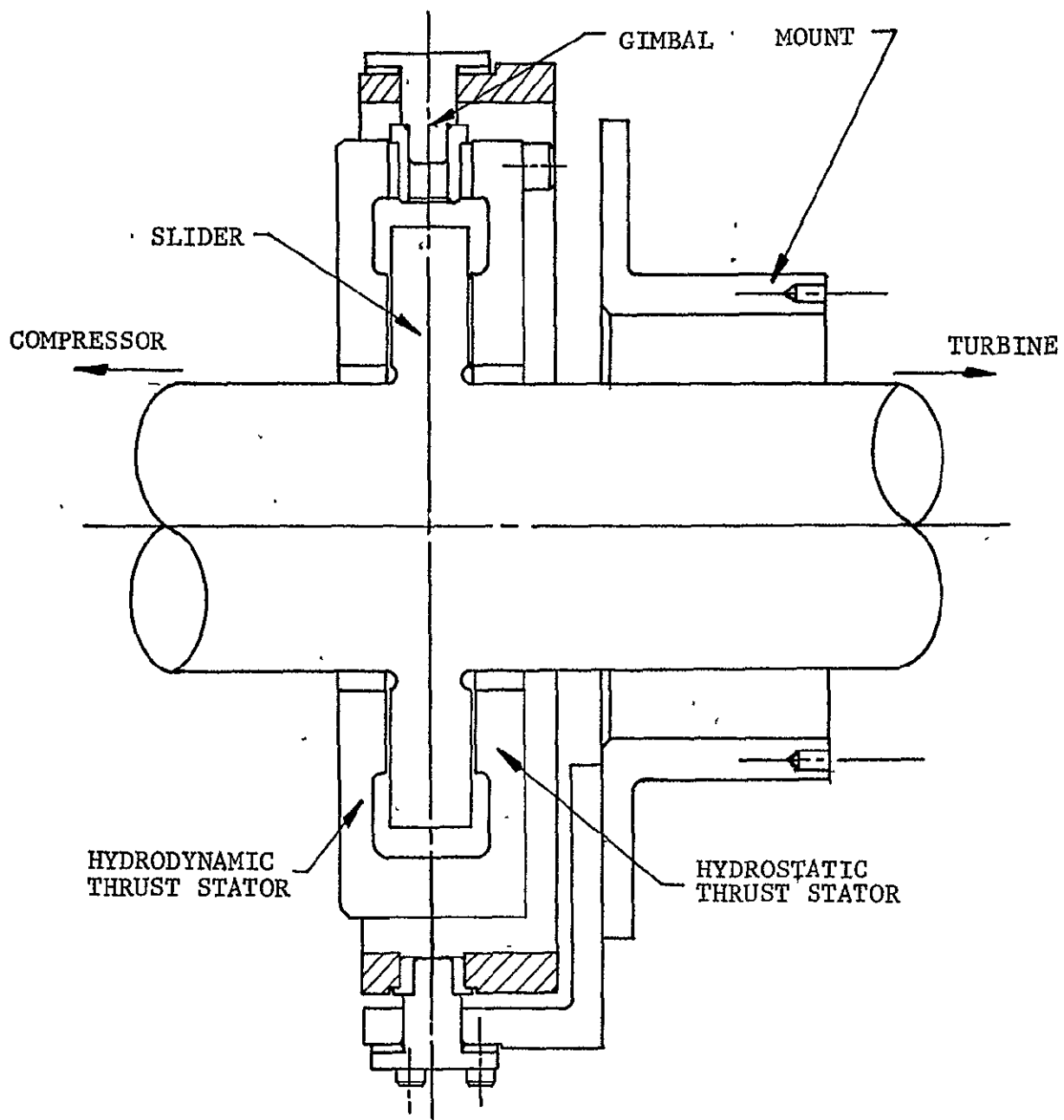
- (d) The bearing is to use argon as the lubricant, with the temperature in the range of 200°F to 800°F.
- (e) Ambient pressure range in the bearing cavity is assumed to vary from 5 psia to 12.0 psia.

Figure 17 shows the double-sided thrust bearing design. The rotor slider is enclosed by two elements; the hydrodynamic bearing is on the compressor side, and the hydrostatic bearing is on the turbine side. The two thrust stators are tied together and supported by a gimbal that allows rotation of the stators in the plane of the slider. Figures 18 and 19 schematically show the hydrostatic and hydrodynamic thrust stators. The hydrostatic stator incorporates six orifices for the hydrostatic lift-off and operation during "reverse" thrust conditions (prior to self-sustained speeds). To facilitate start-up, four hydrostatic orificed pads have been incorporated in the hydrodynamic bearing surface to ensure an adequate operating clearance between the slider and the hydrodynamic bearing, thus precluding rubbing of these bearing surfaces.

By use of a computer program, studies were conducted on the normal thrust bearing to determine an optimum hydrodynamic bearing size based on load capacity and thrust bearing friction at the design operating conditions. Results of this analysis indicated that the thrust bearing surfaces should have an outside radius of 1.62 inches and an inside radius of 0.95 inch. Figures 20, 21, and 22 show the hydrostatic and hydrodynamic performance of the thrust bearings.



AIRESEARCH MANUFACTURING COMPANY OF ARIZONA
A DIVISION OF THE GARRETT CORPORATION
PHOENIX, ARIZONA



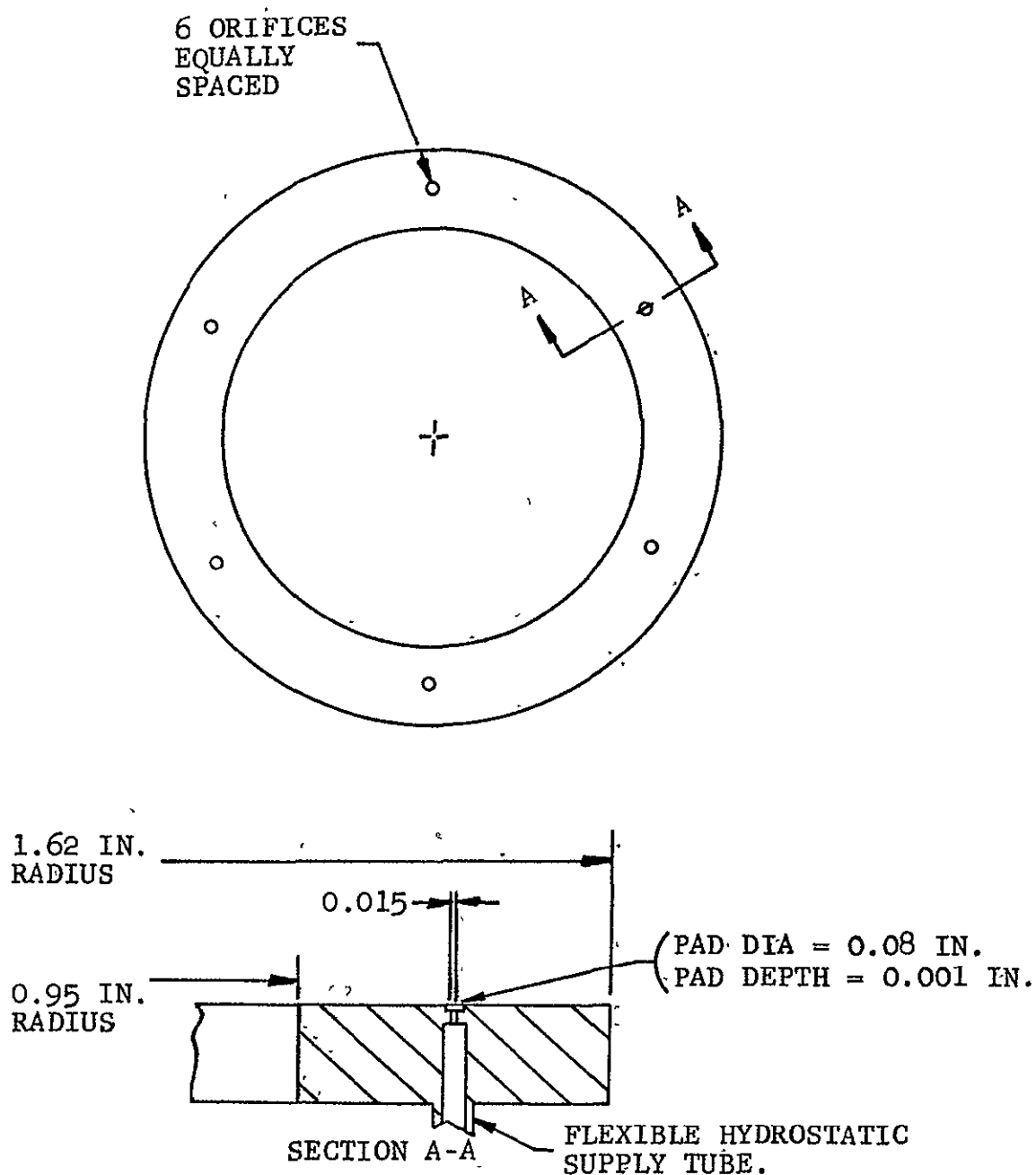
A32550

THRUST BEARING AND GIMBAL ASSEMBLY

FIGURE 17



AIRESEARCH MANUFACTURING COMPANY OF ARIZONA
A DIVISION OF THE GARRETT CORPORATION
PHOENIX, ARIZONA



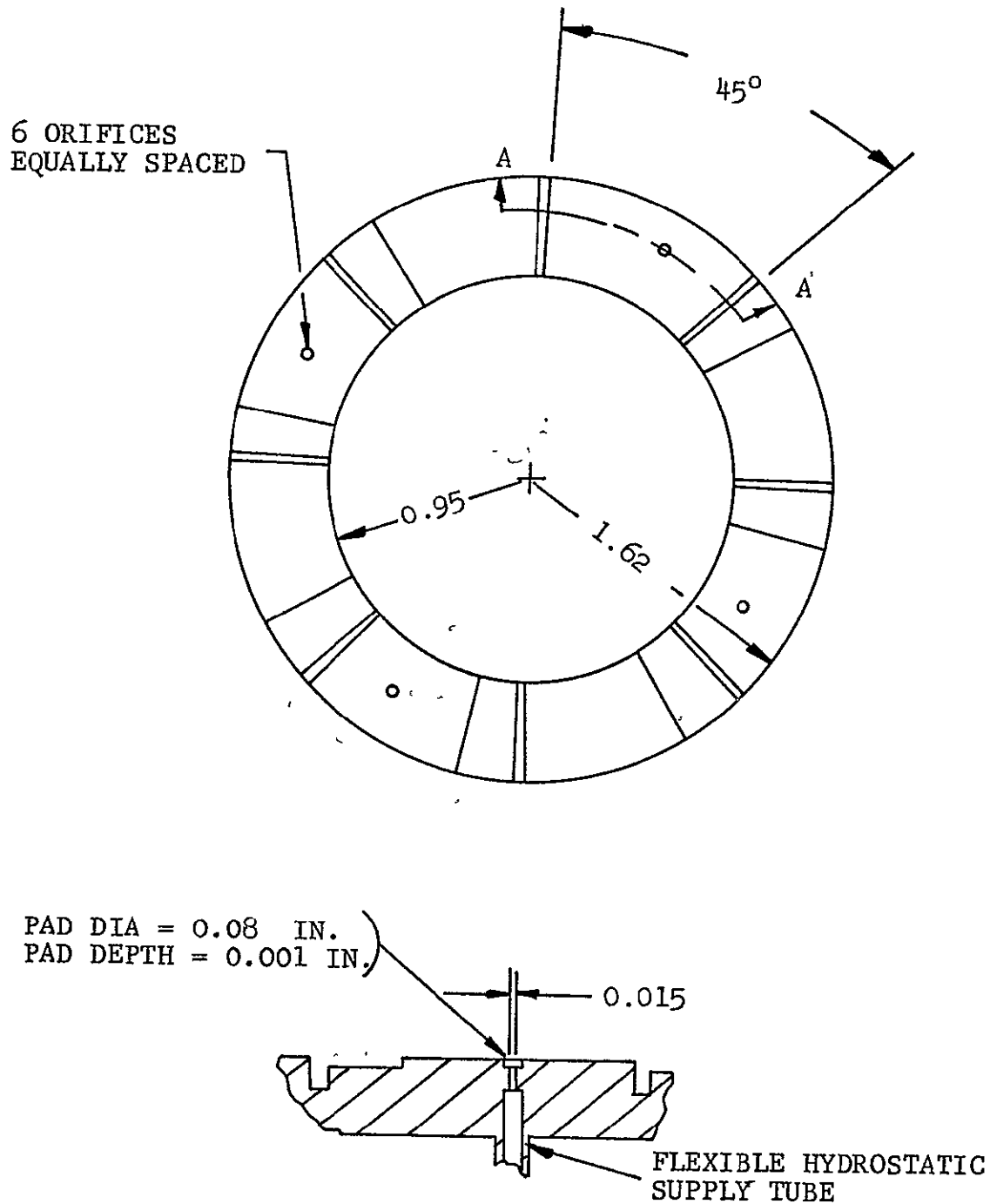
A32569

HYDROSTATIC THRUST BEARING FOR
NASA GAS GENERATOR

FIGURE 18



AIRESEARCH MANUFACTURING COMPANY OF ARIZONA
A DIVISION OF THE GARRETT CORPORATION
PHOENIX, ARIZONA



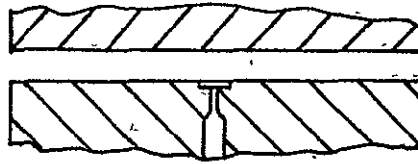
A32570

HYDRODYNAMIC THRUST BEARING FOR NASA GAS GENERATOR

FIGURE 19



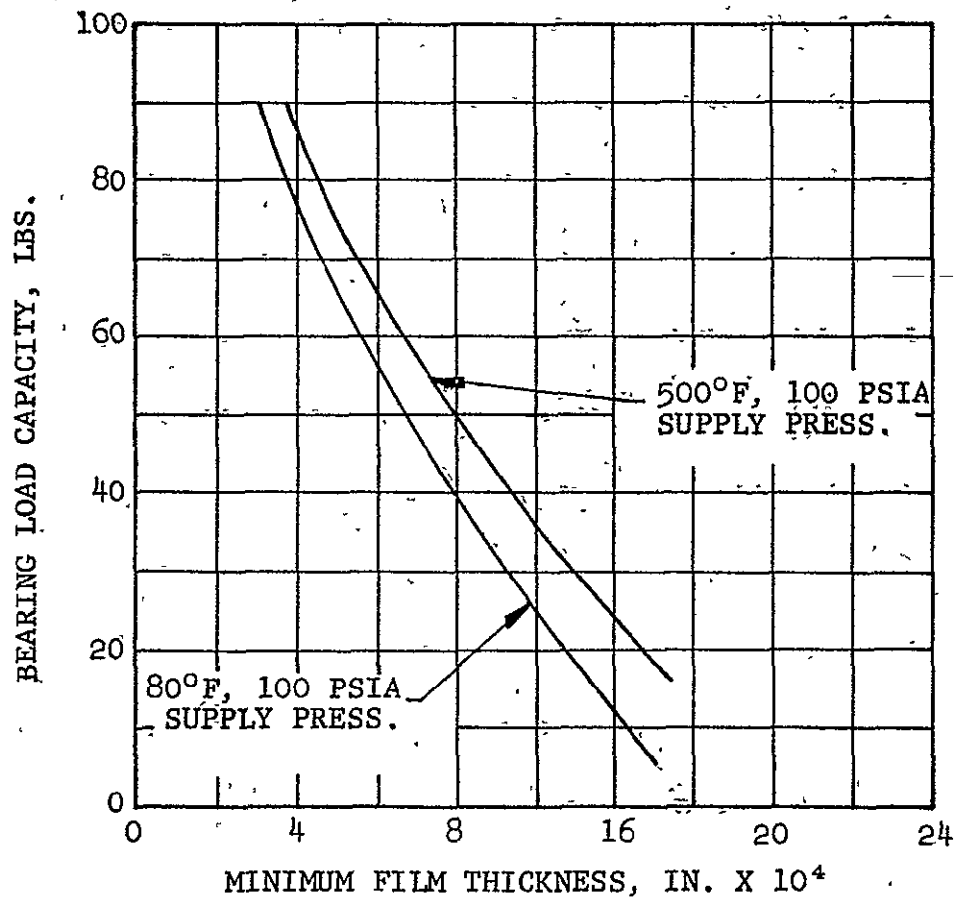
AIRESEARCH MANUFACTURING COMPANY OF ARIZONA
A DIVISION OF THE GARRETT CORPORATION
PHOENIX ARIZONA



HYDROSTATIC STATOR CROSS-SECTION

BEARING CONFIGURATION.

1. OUTER RADIUS = 1.62 INCHES
2. INNER RADIUS = 0.95 INCH
3. AMBIENT PRESSURE = 12 PSIA
4. ARGON LUBRICANT

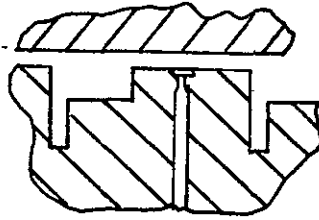


HYDROSTATIC THRUST BEARING LOAD CAPACITY

FIGURE 20



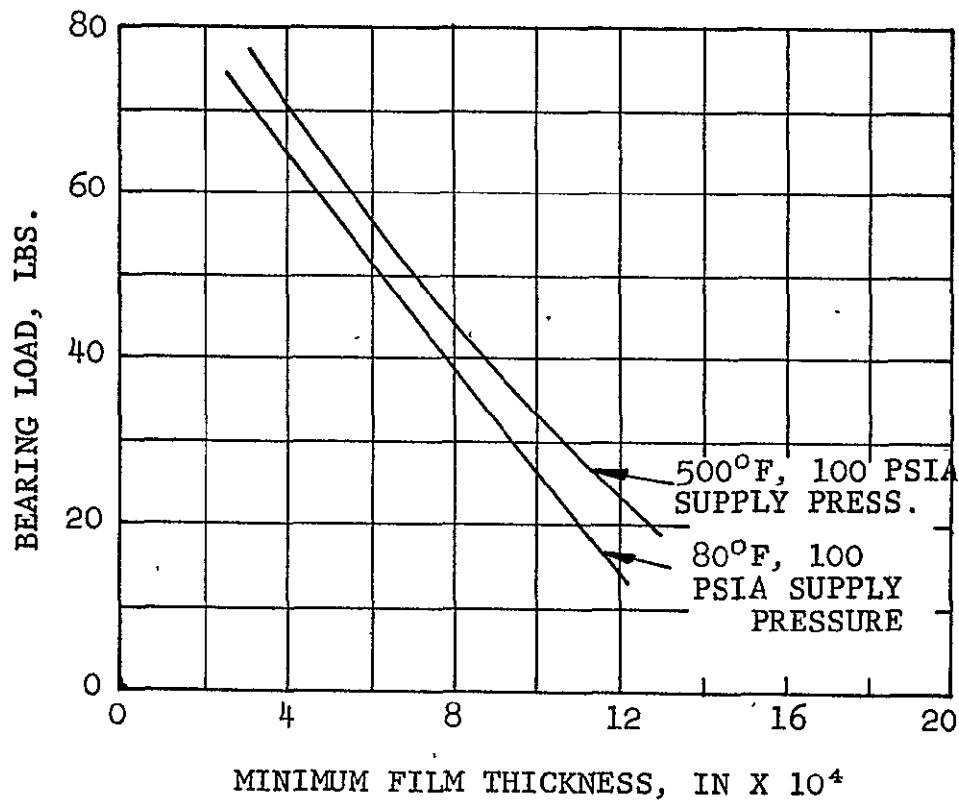
AIRESEARCH MANUFACTURING COMPANY OF ARIZONA
A DIVISION OF THE GARRETT CORPORATION
PHOENIX, ARIZONA



HYDRODYNAMIC THRUST BEARING CROSS-SECTION

BEARING CONFIGURATION

1. OUTER BEARING RADIUS = 1.62 INCHES
2. INNER BEARING RADIUS = 0.95 INCHES
3. AMBIENT PRESSURE = 12 PSIA
4. FOUR OF 8 PADS PROVIDED WITH EXTERNAL PRESSURIZATION ORIFICES
5. ARGON LUBRICANT



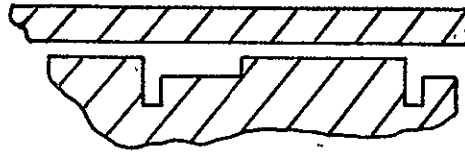
HYDRODYNAMIC THRUST BEARING,
HYDROSTATIC LOAD CAPACITY

A32566

FIGURE 21



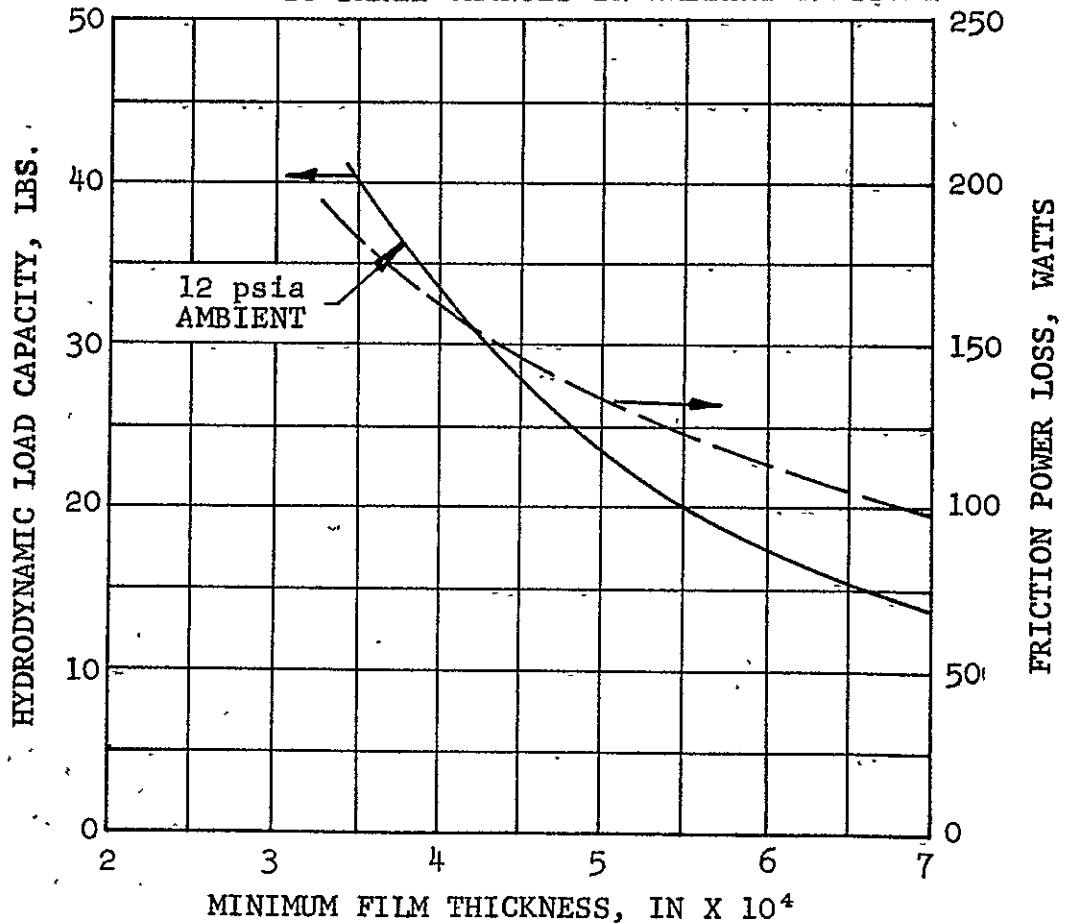
AIRESEARCH, MANUFACTURING COMPANY OF ARIZONA
A DIVISION OF THE GARRETT CORPORATION
PHOENIX, ARIZONA



HYDRODYNAMIC STATOR, CROSS-SECTION

BEARING CONFIGURATION

1. ARGON LUBRICANT, 500°F, $\mu = 5 \times 10^{-8}$
2. SHAFT SPEED = 38,500 RPM
3. THRUST SURFACES ASSUMED PLANE, NO THERMAL DISTORTION CONSIDERED
4. OUTSIDE PAD RADIUS = 1.62 INCHES
INSIDE PAD RADIUS = 0.95 INCH
5. FRICTION POWER GENERATED IS INSENSITIVE TO SMALL CHANGES IN AMBIENT PRESSURE



NASA GAS GENERATOR HYDRODYNAMIC
THRUST BEARING PERFORMANCE

A32567

FIGURE 22



3.5.2 Thrust Bearing Tests

3.5.2.1 Thrust Bearing Integration Test Rig

The integration of the thrust-bearing/gimbal-mount assembly and journal bearings was accomplished on a bearing test rig that very closely simulates the final gas generator design. The rig consists of a housing, bearing carriers, journal gas bearings, and shaft that exactly duplicate the final gas generator configurations. Dummy masses are included on the shaft to simulate the masses of the journal and turbine wheels. An air turbine motor is utilized to drive the shaft. Loading of the thrust bearing can be varied by introducing a positive or negative pressure in a cavity under the shaft. The test-rig instrumentation is similar to the final gas-generator instrumentation. Figure 23 shows the bearing test rig as installed in the bearing test facility.

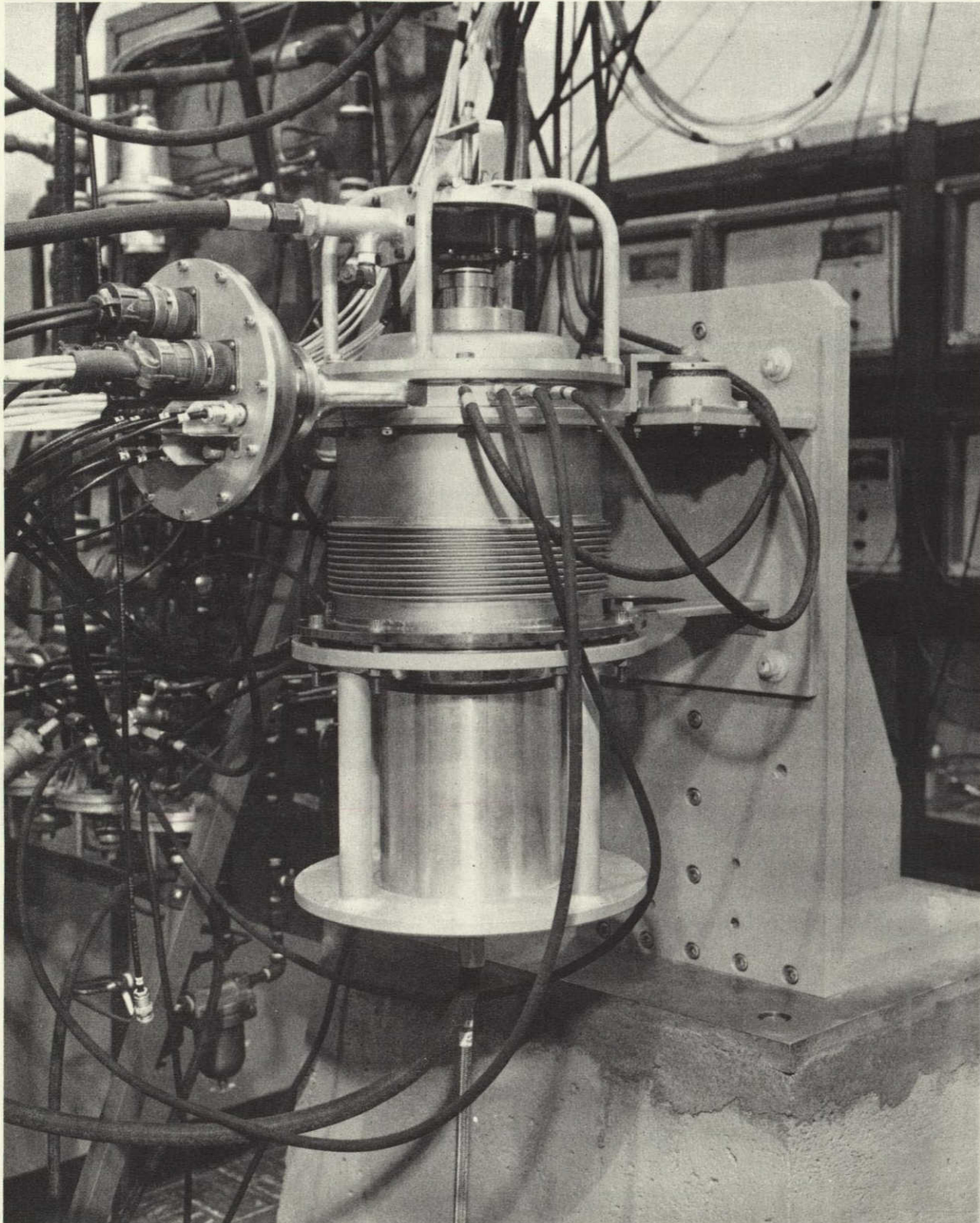
3.5.2.2 Thrust Bearing Integration Tests

Various gimbal configurations were tested with varying degrees of success. The configuration that ultimately proved to be the most successful -- and the one which was incorporated into the gas generator -- utilized two pairs of pin-in-socket pivots, each pair located in the mean plane of the thrust surfaces.

Stable operation was obtained on both thrust surfaces, and thrust transfers between the thrust surfaces were very satisfactory. Loads up to 80 pounds were achieved on the reverse thrust bearing (hydrostatically). Loads from 0 to 35 pounds and 5 to 28 pounds were achieved on the normal thrust surface operating hydrostatically and hydrodynamically, respectively. These tests, undertaken on the frame test rig, were considered to have demonstrated that integration of the journal and thrust bearings had been achieved.



AIRESEARCH MANUFACTURING COMPANY OF ARIZONA
A DIVISION OF THE GARRETT CORPORATION
PHOENIX, ARIZONA



COMPANY-SPONSORED BEARING TEST RIG
FIGURE 23



3.6 Thermal Analysis

A digital computer thermal analysis program is the only economical means by which an integrated housing, rotating group/bearing assembly can be analyzed from a thermal standpoint to determine:

- (a) Operating temperatures throughout the machine
- (b) Relative parts movements
- (c) Thermally induced stresses
- (d) Material requirements
- (e) Cooling flows if required
- (f) Bearing operating requirements

Thermal analyses are conducted at AiResearch with digital computer programs by mathematically defining the complete turbo-machine together with the inlet and discharge design conditions, bearing frictional power losses, bearing hydrostatic gas supply flow, and various cooling flow rates, etc. Information necessary for the thermal analyses is the basic machine configuration as defined by the preliminary layout; the wheel configuration as defined by the stress and aerodynamic analyses; and the bearing spacing, bearing initial configuration, and power losses as defined by the dynamic and bearing analyses.

Materials for the various component parts are selected from an initial thermal analysis, and cooling flows are selected, if required. The analysis is then performed again with the configuration



modifications until bearing temperatures and thermal gradients are reduced to within acceptable bearing design, stress, and growth limitations.

A thermal analysis by use of the method discussed above was conducted on the basic gas generator and the steady-state temperature results are shown in Figure 24.

Figure 25 illustrates the heat flow paths and the quantities of heat energy being transferred through the rotor and associated parts in the gas generator. It may be noted that approximately 50 percent of the heat input to the shaft is convected away by the cooling gas. The remaining 50 percent is conducted to the impeller where it is convected to the compressor through-flow. Figure 26 presents the gas generator's shaft temperature distribution at steady state and at three values of time during the start-up transient.

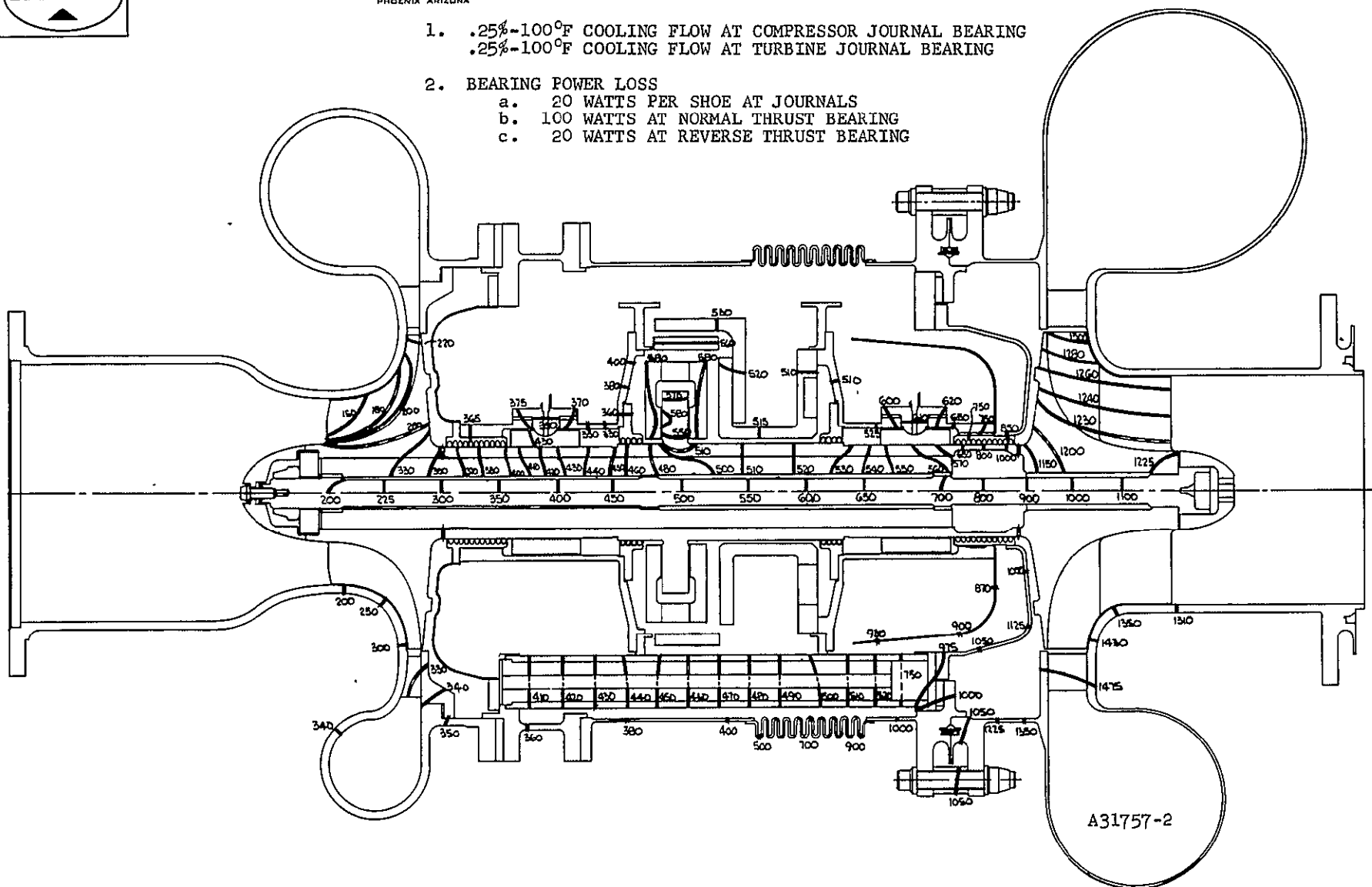
Figure 27 presents the transient and steady-state compressor and turbine axial shroud clearances. With initial shroud clearance settings of 0.012 and 0.014 inch for the compressor and turbine, respectively, steady-state running clearances of approximately 0.010 inch will be achieved. During transient operation the shroud clearances open up to a maximum of 0.017 and 0.020 inch for the compressor and turbine, respectively.

Figure 28 presents the temperature distribution at the bearing shoe and the adjacent journal area, for the compressor and turbine bearings, respectively, at steady state and at three time points during the transient. These illustrate how closely in temperature level the shoes and the journals follow each other and that they are relatively free from coning and crowning.



AIRESEARCH MANUFACTURING COMPANY OF ARIZONA
A DIVISION OF THE GARRETT CORPORATION
PHOENIX, ARIZONA

1. .25%-100°F COOLING FLOW AT COMPRESSOR JOURNAL BEARING
.25%-100°F COOLING FLOW AT TURBINE JOURNAL BEARING
2. BEARING POWER LOSS
 - a. 20 WATTS PER SHOE AT JOURNALS
 - b. 100 WATTS AT NORMAL THRUST BEARING
 - c. 20 WATTS AT REVERSE THRUST BEARING

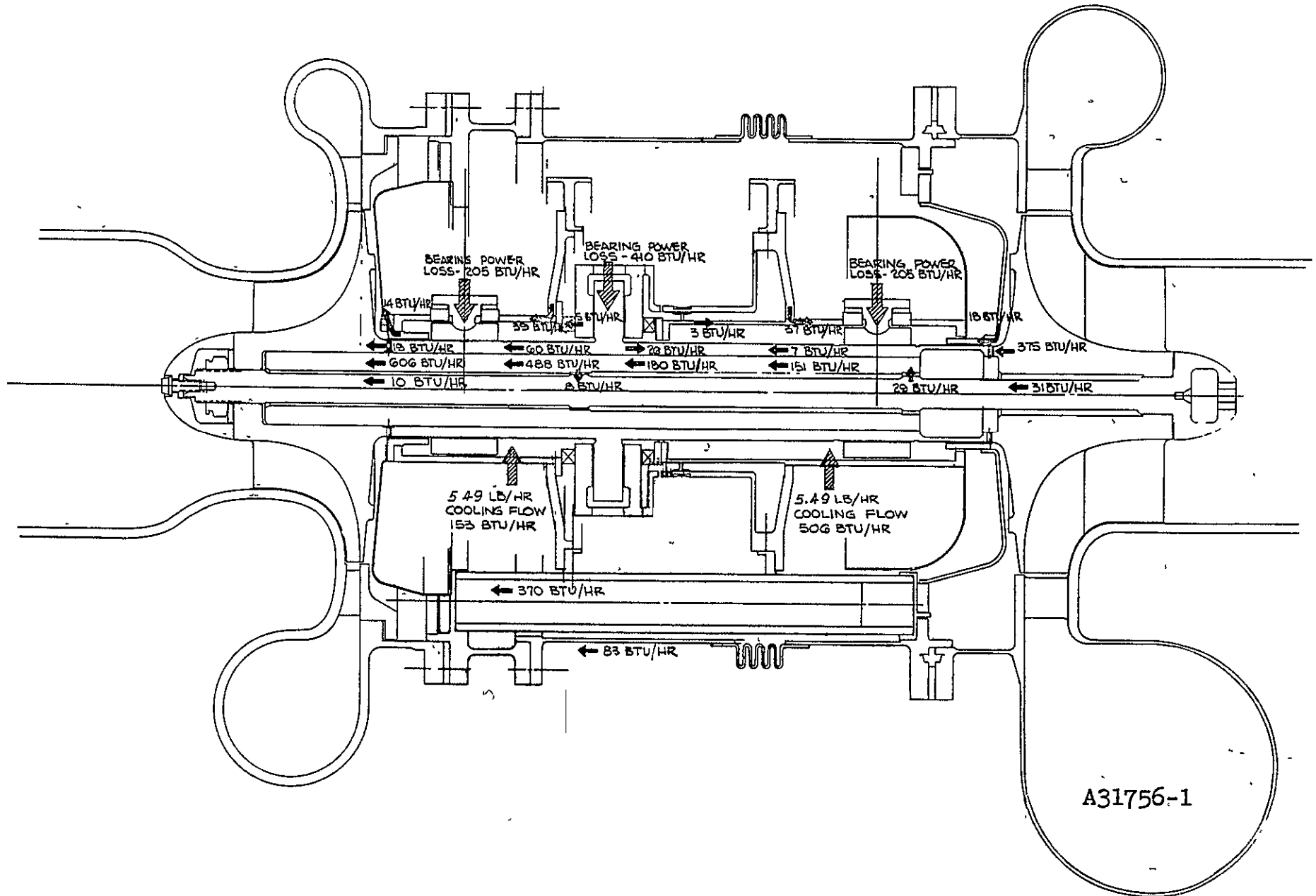


STEADY STATE TEMPERATURE DISTRIBUTION

FIGURE 24



AIRESEARCH MANUFACTURING COMPANY
A DIVISION OF THE GARRETT CORPORATION
PHOENIX, ARIZONA

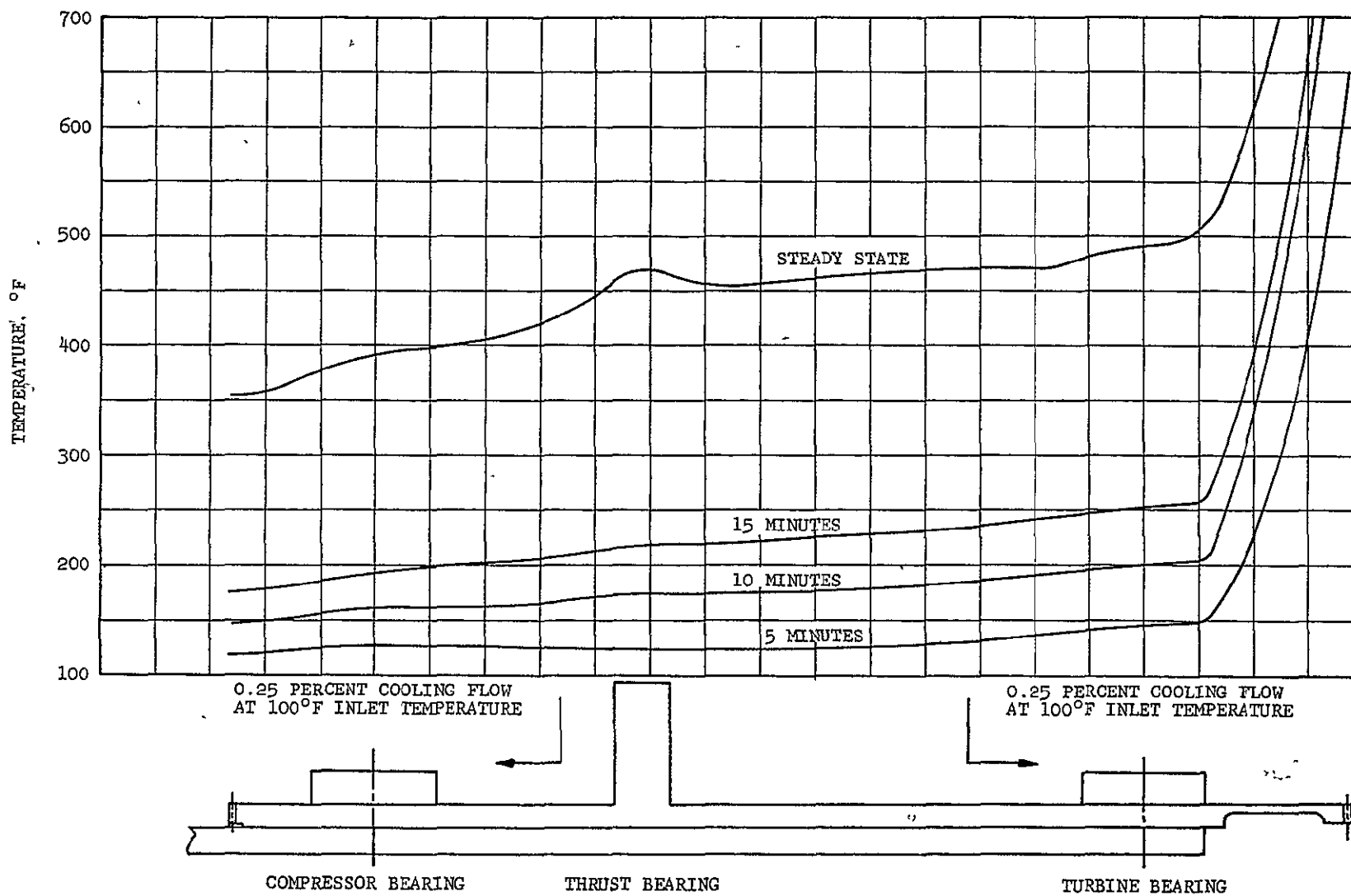


A31756-1

HEAT FLOW DIAGRAM
FIGURE 25



AIRESEARCH MANUFACTURING COMPANY OF ARIZONA
A DIVISION OF THE GARRETT CORPORATION
PHOENIX, ARIZONA

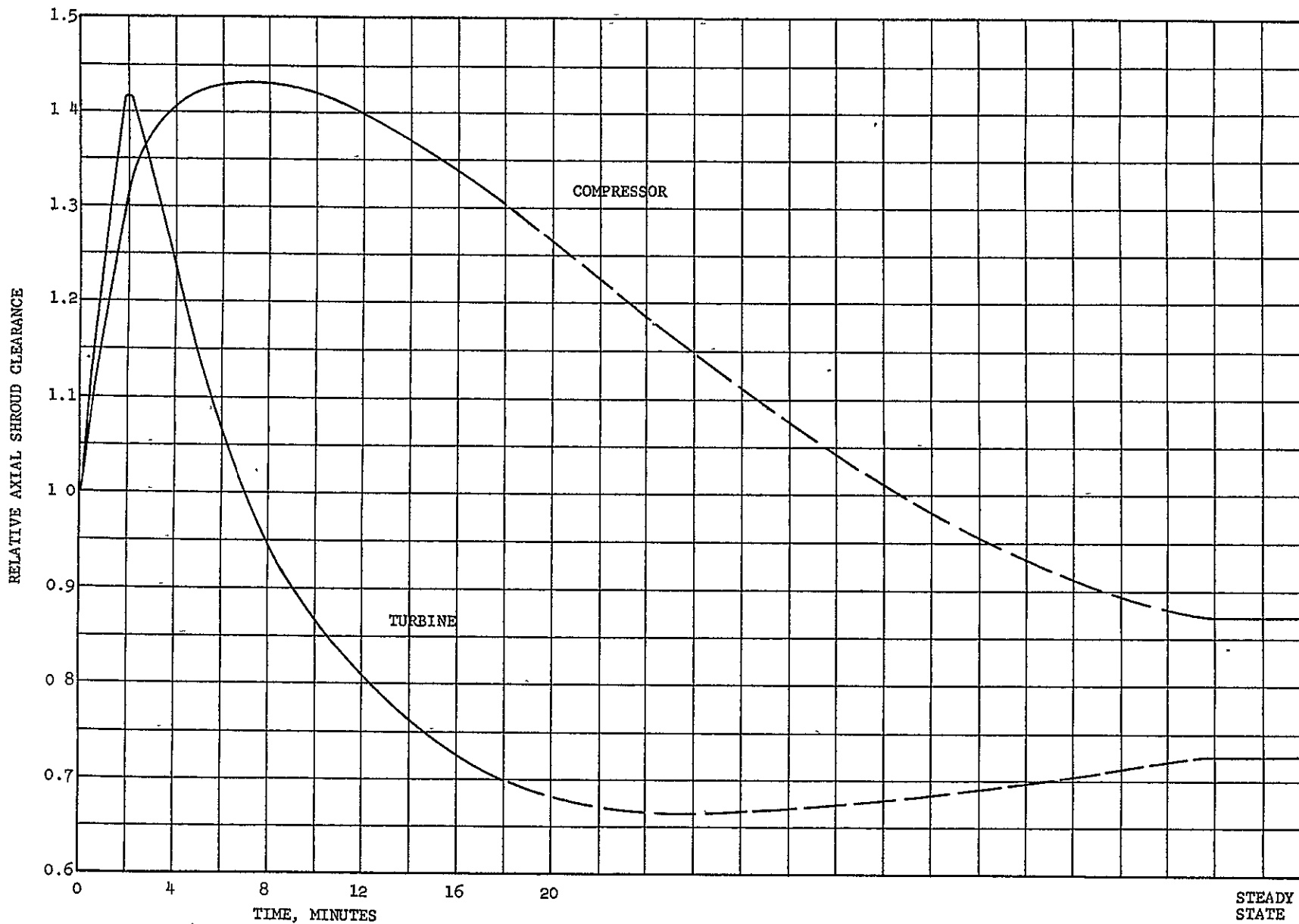


SHAFT TEMPERATURES AT DIFFERENT VALUES OF TIME

FIGURE 26



AIRESEARCH MANUFACTURING COMPANY OF ARIZONA
A DIVISION OF THE GARRETT CORPORATION
PHOENIX, ARIZONA



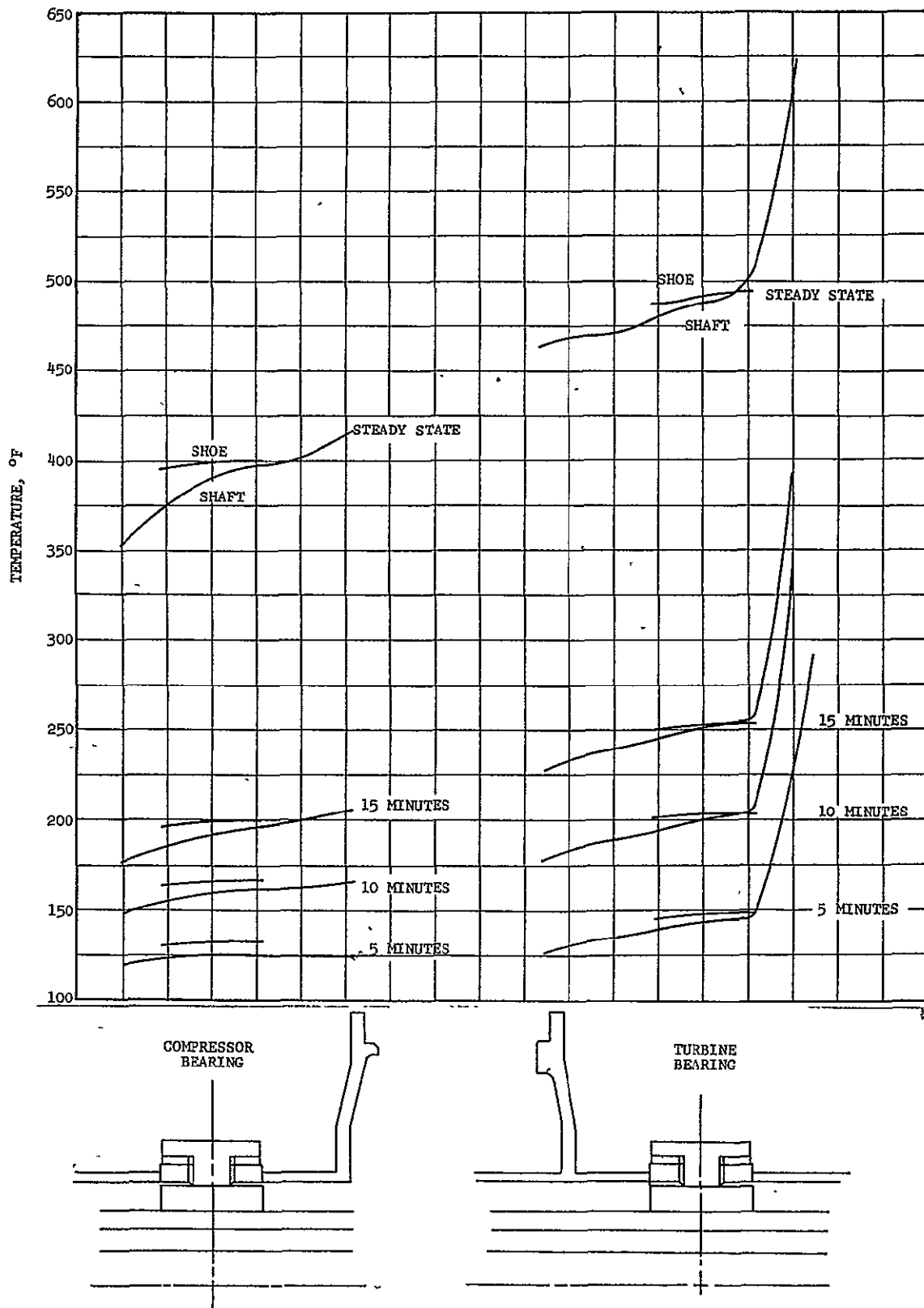
TRANSIENT AND STEADY-STATE COMPRESSOR
AND TURBINE SHROUD CLEARANCE

FIGURE 27



AIRESEARCH MANUFACTURING COMPANY OF ARIZONA

A DIVISION OF THE GARRETT CORPORATION
PHOENIX, ARIZONA



COMPRESSOR AND TURBINE BEARING AND SHAFT TEMPERATURES
AT DIFFERENT VALUES OF TIME

A31869

FIGURE 28



Figure 29 presents the transient and steady-state relative movement between the bearing journals and the bearing carriers. Relative parts movement for the gas generator is in the order of 0.001 inch and must be accommodated by the resiliently mounted gas bearings.

Figure 30 presents the transient and steady-state position of the mass center lines of the journals relative to the geometric center lines of the scroll shrouds. The geometric center line of the shaft was initially displaced 0.002 inch relative to the geometric center line of the shrouds. Hence, the gas generator must be assembled with the shaft intentionally displaced approximately 0.002 inch, relative to the shrouds, in the direction opposite to the spring mount. This will give, as Figure 30 indicates, a maximum shift of centers of 0.002 and a steady-state running position that is nearly centered in the shrouds.

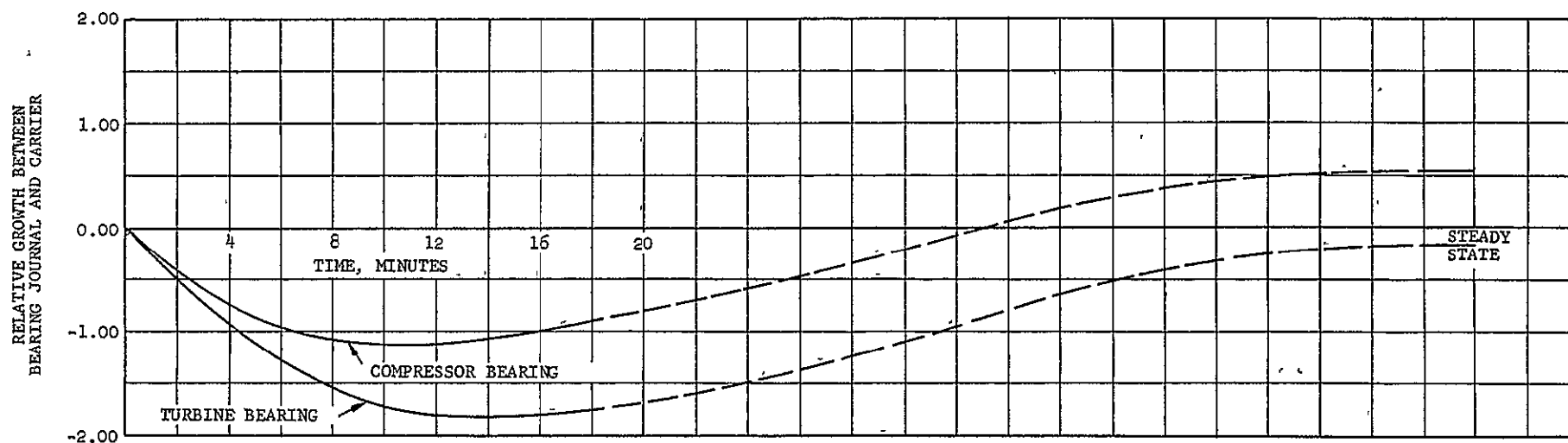
Figure 31 shows the transient and steady-state loads imposed on the compressor and turbine journal bearings. Effects considered in the load analysis were centrifugal growth, unbalance, relative thermal growth, and film thickness variation with load and ambient temperatures. An initial preload of 10 pounds and a rotor unbalance of 0.0002 c.g. eccentricity were assumed, and a spring rate of 1,500 pounds per inch was selected for the single, flexibly mounted bearing. For the thermally induced changes predicted by the gas generator thermal analysis, reasonable bearing loads and load transients exist with the gas generator design.

Figure 32 presents transient and steady-state temperatures for several representative points located throughout the gas generator. Points such as the turbine nozzle, the turbine wheel hub, and the diffuser vane react rather quickly, reaching their steady-state temperatures in a matter of a few minutes. Other



AIRESEARCH MANUFACTURING COMPANY OF ARIZONA
A DIVISION OF THE GARRETT CORPORATION
PHOENIX, ARIZONA

NOTE: NEGATIVE ORDINATE INDICATES GROWTH OF JOURNAL INTO CARRIER.



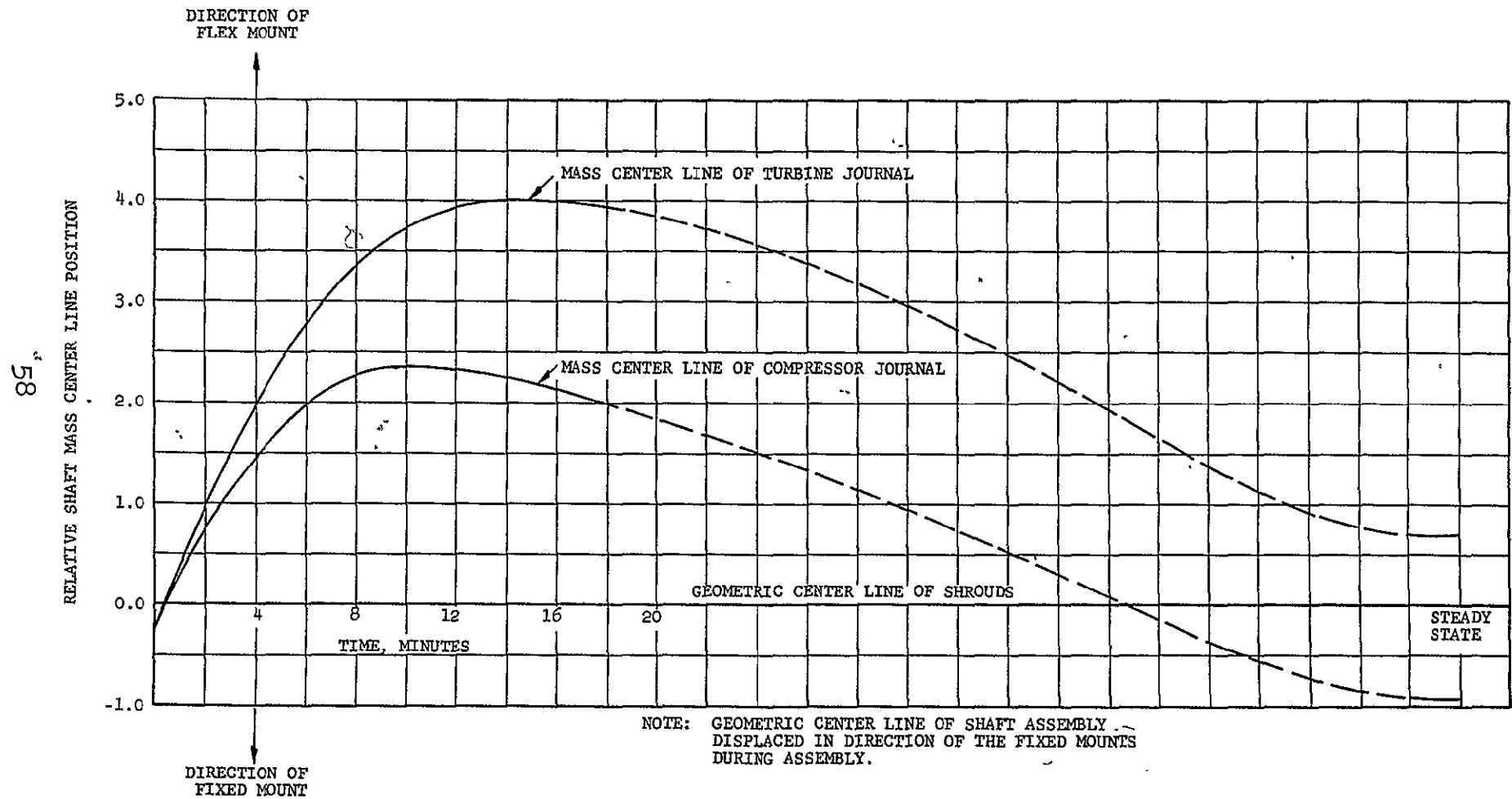
TRANSIENT AND STEADY-STATE BEARING JOURNAL
TO CARRIER RELATIVE EXPANSION

A31745-1

FIGURE 29



AIRESEARCH MANUFACTURING COMPANY OF ARIZONA
A DIVISION OF THE GARRETT CORPORATION
PHOENIX ARIZONA



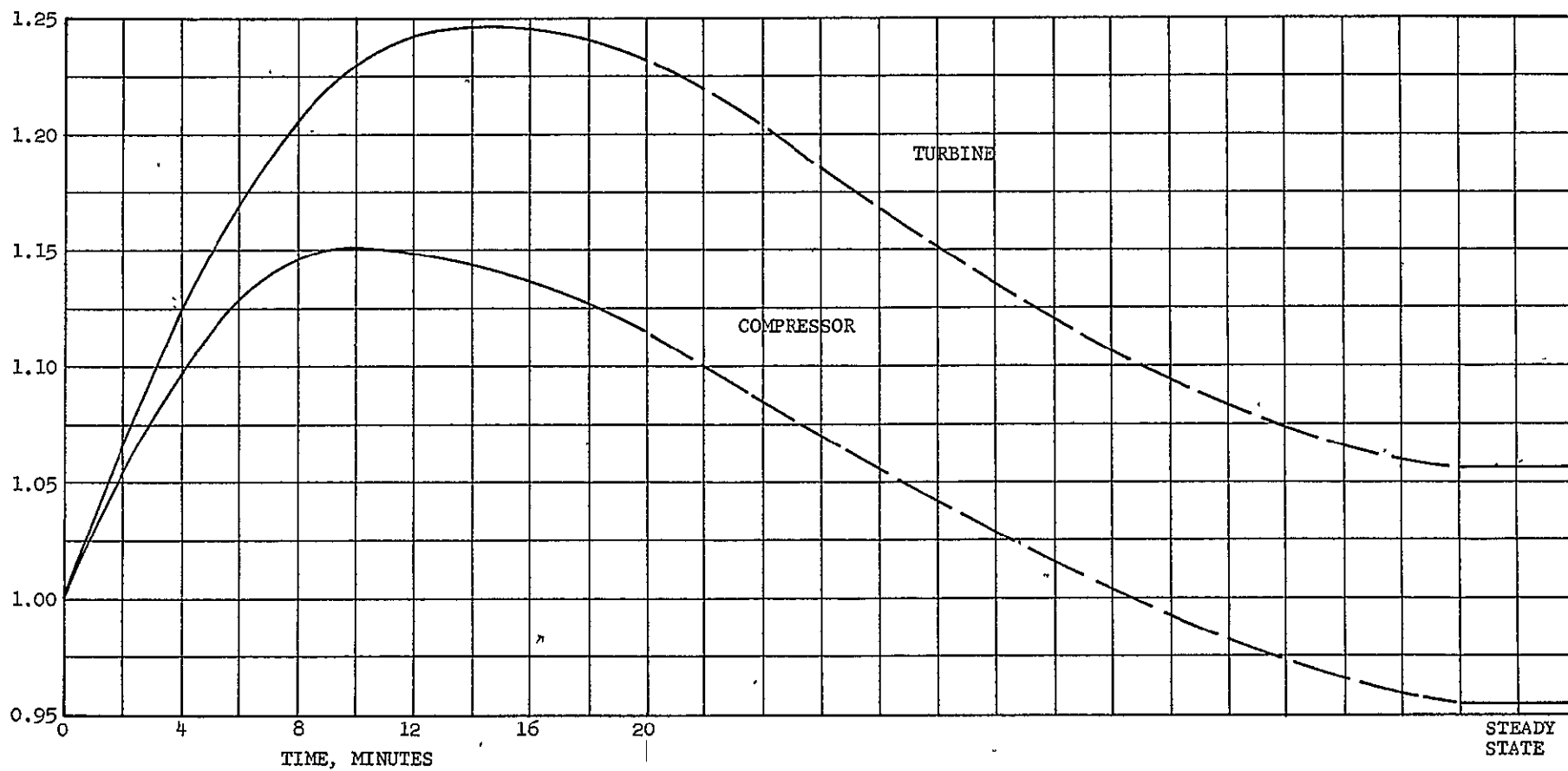
TRANSIENT AND STEADY STATE OF THE
MASS CENTER LINE OF THE SHAFT

FIGURE 30

A31746-1



69
RELATIVE JOURNAL BEARING LOAD



TRANSIENT AND STEADY-STATE JOURNAL BEARING LOADS

A31747-1

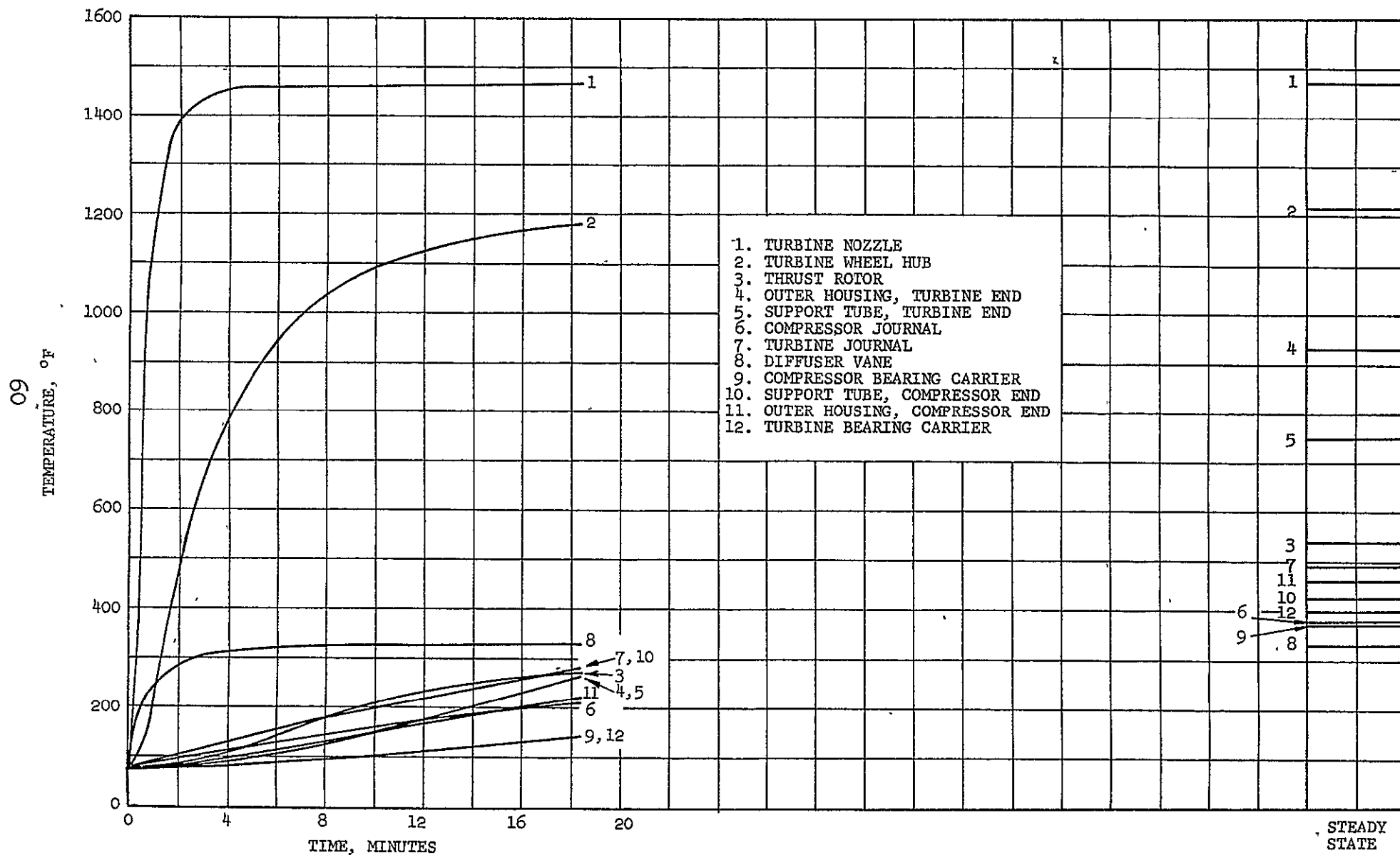
FIGURE 31



AIRESEARCH MANUFACTURING COMPANY OF ARIZONA

A DIVISION OF THE GARRETT CORPORATION

PHOENIX, ARIZONA



TRANSIENT AND STEADY-STATE TEMPERATURES OF REPRESENTATIVE POINTS IN THE GAS GENERATOR

FIGURE 32



points such as those in the support structure and bearing carrier take an unpredictably long time to reach steady state.

During the course of the thermal analysis, several combinations of coolant flow rates and coolant initial temperatures were investigated in an attempt to determine if there were any optimum combination. Figure 33 presents four of the many schemes investigated. These four, shown in dashed lines, are compared to the final design, indicated by the solid line, all using identical hardware. The accompanying table lists all the particulars of each scheme and presents the steady-state values of relative expansion between the bearing carrier and the journals for each scheme. In all instances the relative expansions are well within the capability limits of the gas generator final design. It appears that there should be no concern about choosing a cooling arrangement for its own sake, because of the relative freedom to base the selection on overall system performance and/or complexity.

Cooling scheme "A" was therefore selected, since, with the labyrinth seals at their minimum clearance, the seals could be opened up to increase the cooling flow during evaluation testing by the NASA.

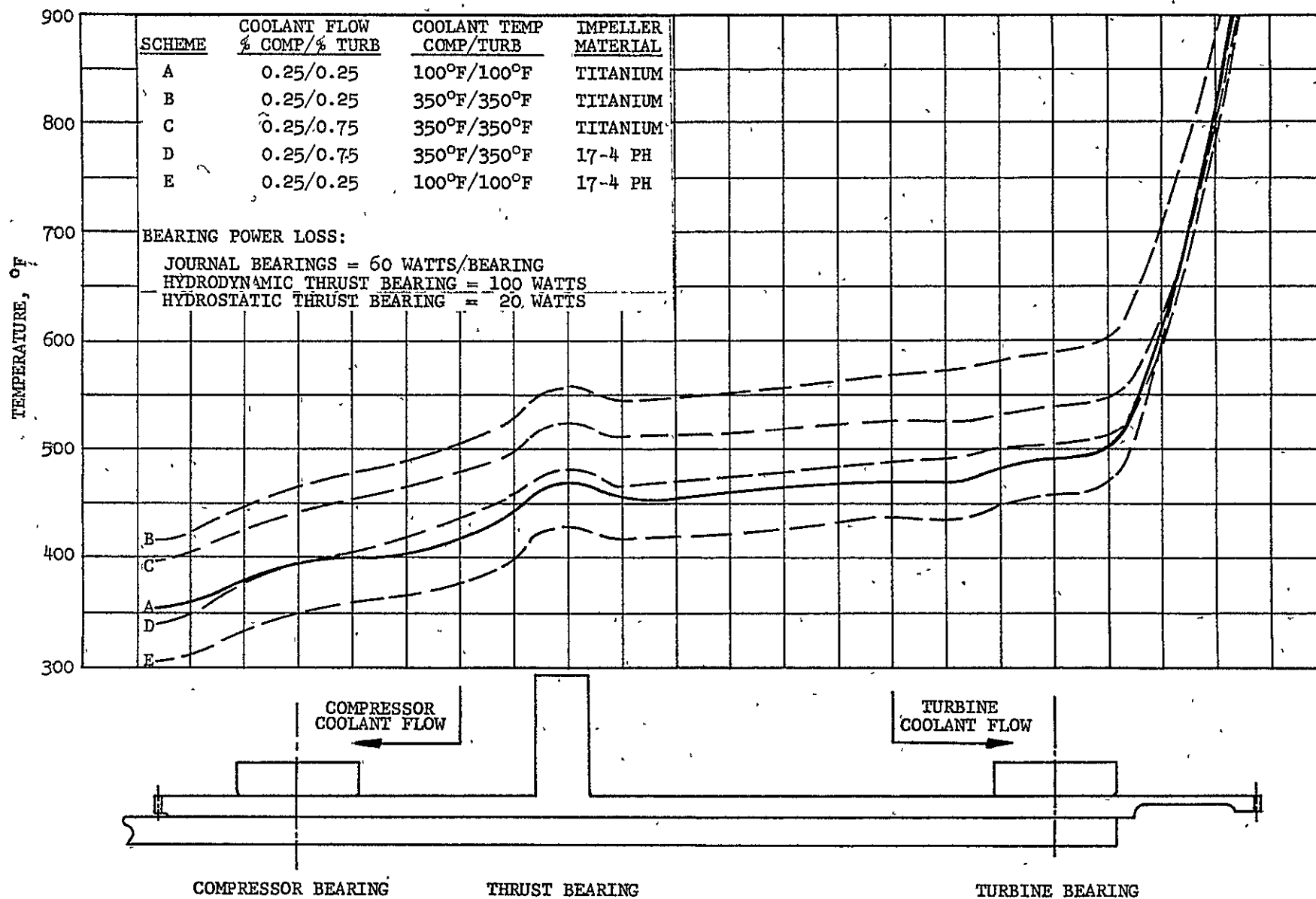
3.7 Shock and Vibration Analysis

For the shock and vibration analysis of the gas generator, the system analyzed consisted of three bodies defined as follows:

Body 1 - Large mass representative of the
 launch vehicle structure

Body 2 - Gas-generator housing

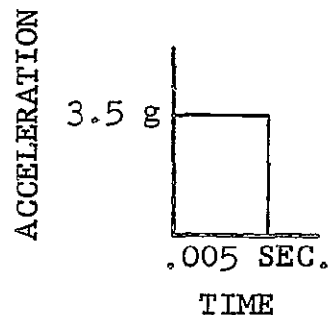
Body 3 - Gas-generator rotating assembly



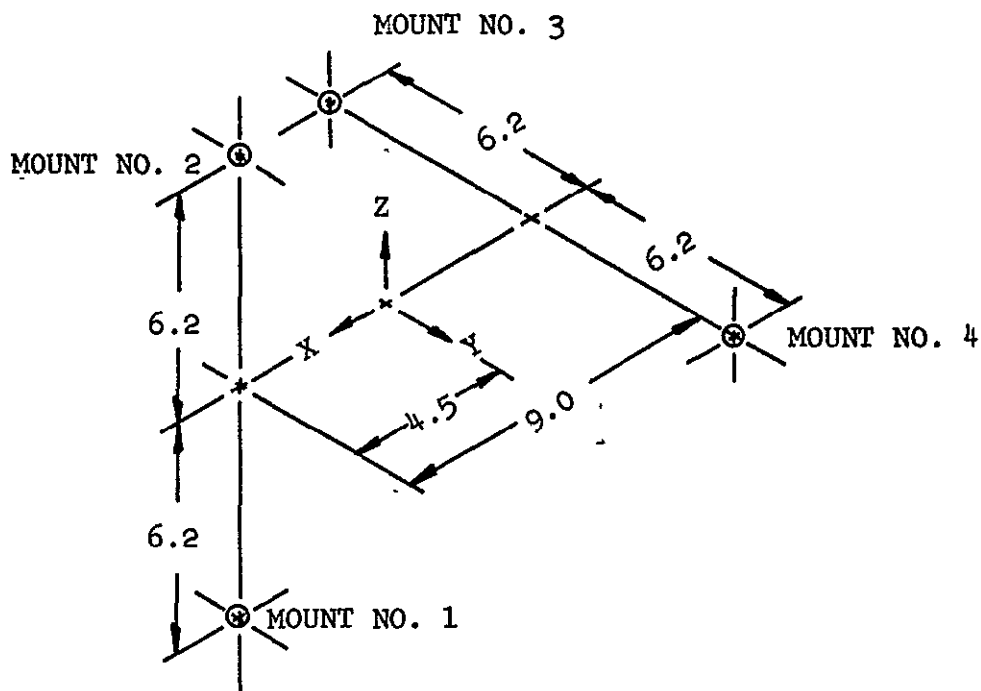


The gas generator is isolated from the launch vehicle so that the acceleration and displacement imposed are within acceptable limits.

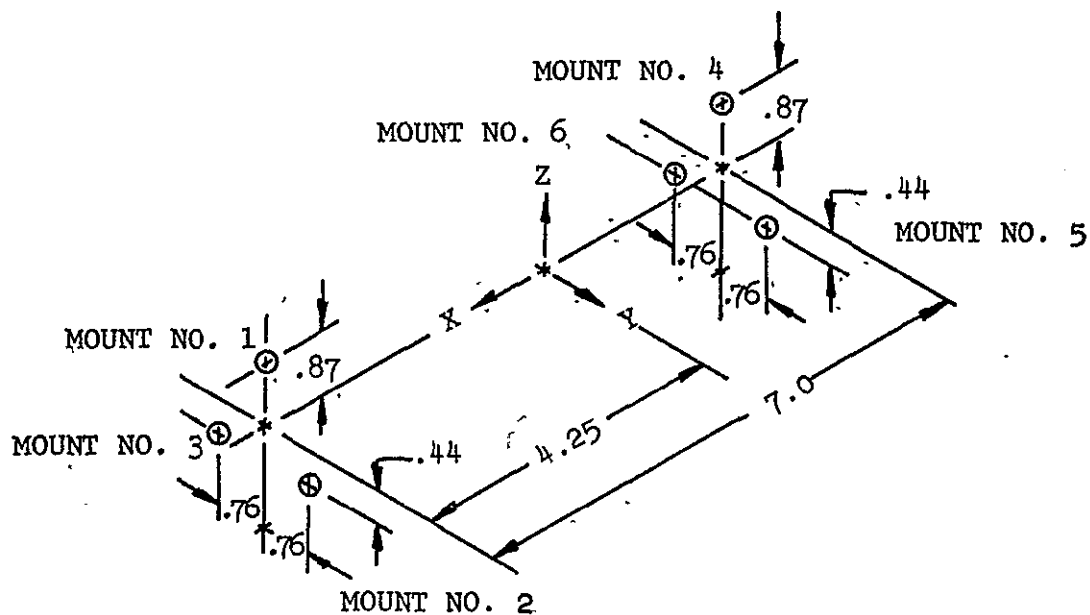
Figures 34 and 35 are schematics of the mount locations used in the design analysis, and Table 2 shows the spring rates of the mounts for each body during launch. It should be noted that, to have a complete analysis of the system, additional information would be required concerning the ducts to be attached to the gas generator. During launch the minimum vibratory frequency to be isolated is 61 cps. The gas-generator mounts should be designed to have a spring rate that will provide a natural frequency below 16 cps. This places the system resonant condition in a region of impressed frequency of relatively low input acceleration. By using the spring rates listed in Table 2, the natural frequency of the system is found to be 13.7 cps. This result is for an impressed acceleration in the Z direction shown below.



With the same impressed acceleration in both the X and the Z directions, the resulting loads are as shown in Tables 3 and 4. Note that the loads for Body 2 (97 pounds in weight) and Body 3 (15.5 pounds in weight) show that the 3.5-g impressed acceleration is effectively isolated. Figure 36 shows a shear and bending moment diagram for the Z direction shock application.



SCHEMATIC OF GAS GENERATOR FRAME
MOUNT POINTS (BODY 2)
FIGURE 34



SCHEMATIC OF GAS GENERATOR SHAFT
MOUNT POINTS (BODY 3)
FIGURE 35



TABLE 2

SUMMARY OF SPRING RATES FOR LAUNCH CONDITION

K = LBS/INCH	BODY 2				BODY 3					
MOUNT	1	2	3	4	1	2	3	4	5	6
SPRING RATE										
K_x	65.0	65.0	650.0	650.0	10^6	0	0	10^6	0	0
K_y	650.0	65.0	65.0	650.0	0	$5(10)^5$	$5(10)^5$	0	$5(10)^5$	$5(10)^5$
K_z	65.0	650.0	650.0	650.0	$3(10)^5$	$5(10)^5$	$5(10)^5$	$3(10)^5$	$5(10)^5$	$5(10)^5$

TABLE 3

SUMMARY OF MAXIMUM FORCES AT MOUNTS
FOR RECTANGULAR SHOCK PULSE IN
X DIRECTION DURING LAUNCH CONDITION

F = LBS	BODY 2				BODY 3					
MOUNT	1	2	3	4	1	2	3	4	5	6
FORCE										
F_x	-48.6	-48.6	-485.6	-485.6	-92.2	0	0	-92.2	0	0
F_y	0	0	0	0	0	0	0	0	0	0
F_z	0	0	0	0	1.1	10.9	10.9	0	-10.6	-10.6

TABLE 4

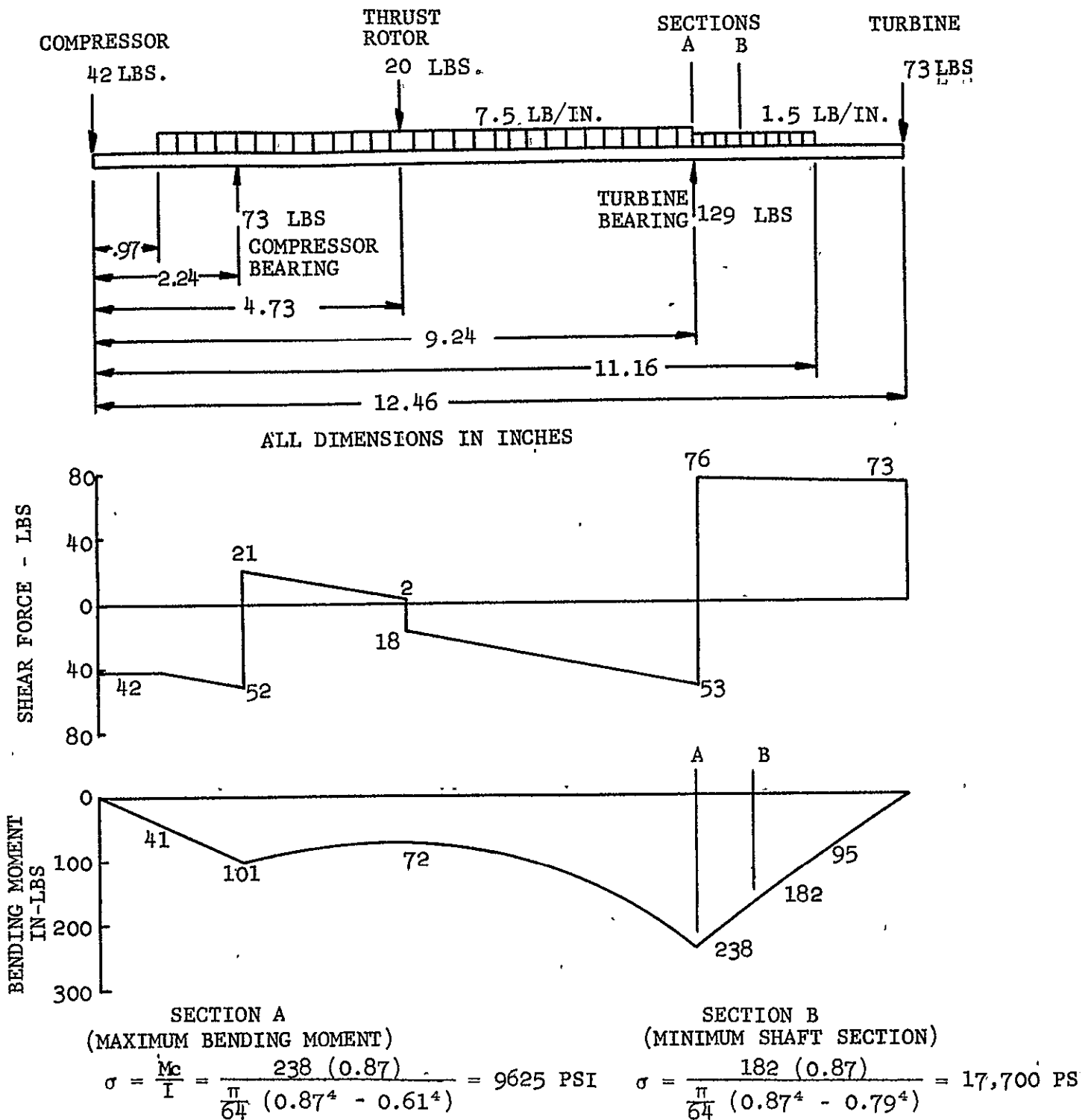
SUMMARY OF MAXIMUM FORCES AT MOUNTS
FOR RECTANGULAR SHOCK PULSE IN
Z DIRECTION DURING LAUNCH CONDITION

F = LBS	BODY 2				BODY 3					
MOUNT	1	2	3	4	1	2	3	4	5	6
FORCE										
F_x	9.9	-9.9	0	0	-.3	0	0	-.3	0	0
F_y	0	0	0	0	0	0	0	0	0	0
F_z	-46.4	-466	-324.4	-324.4	-.2	-36.5	-36.5	-.4	-64.4	-64.4

A32547



AIRESEARCH MANUFACTURING COMPANY OF ARIZONA
A DIVISION OF THE GARRETT CORPORATION
PHOENIX, ARIZONA



LOAD, SHEAR, AND BENDING MOMENT DIAGRAM
FOR RECTANGULAR SHOCK PULSE IN
Z DIRECTION DURING LAUNCH CONDITION

A32548

FIGURE 36



With the mount system that has been designed, the vibration frequencies that can cause the most damage have been isolated. The mount system also effectively reduces the acceleration transmitted through the mounts to the gas generator housing and rotating assembly.

3.8 Instrumentation

During the journal and thrust bearing testing, the instrumentation incorporated in the bearing test rigs (described in Section 3.4) was found to be very useful for setting up the bearings and determining the bearing performance. Bearing and shaft stability was monitored with capacitance probes. Bearing loads were determined and monitored with strain gauges. Thermocouples were utilized to determine bearing temperatures as well as the other instrumentation component temperatures for calibration purposes.

Based on the instrumentation utilized during the bearing testing, the instrumentation required to monitor the operation of the gas generator was analyzed and the requirements were determined as follows:

- (a) Rotor shaft position at both the turbine and compressor journals, with proximity probes in quadrature (four probes required).
- (b) Thrust bearing gas film thickness, with the use of proximity probes on the hydrostatic and hydrodynamic thrust faces (two probes required).
- (c) Journal bearing shoe loads, with strain gauges used on the two flexible diaphragm mounts (eight gauges required).



- (d) Thrust bearing loads, with strain gauges used on the bearing mount (four strain gauges required).
- (e) Thermocouples monitoring temperatures within the machine (54 thermocouples required).
- (f) Shaft speed, with three electronic speed pickups used in the same shaft plane (three required).
- (g) Static pressure taps on the turbine scroll assembly (two required).
- (h) Static pressure taps within the machine (two required for bearing labyrinths and one required for the cavity).

3.8.1 Capacitance Probes

Since commercially available proximity probes were not available, the required probes were purchased from the AiResearch Instrumentation Laboratory.

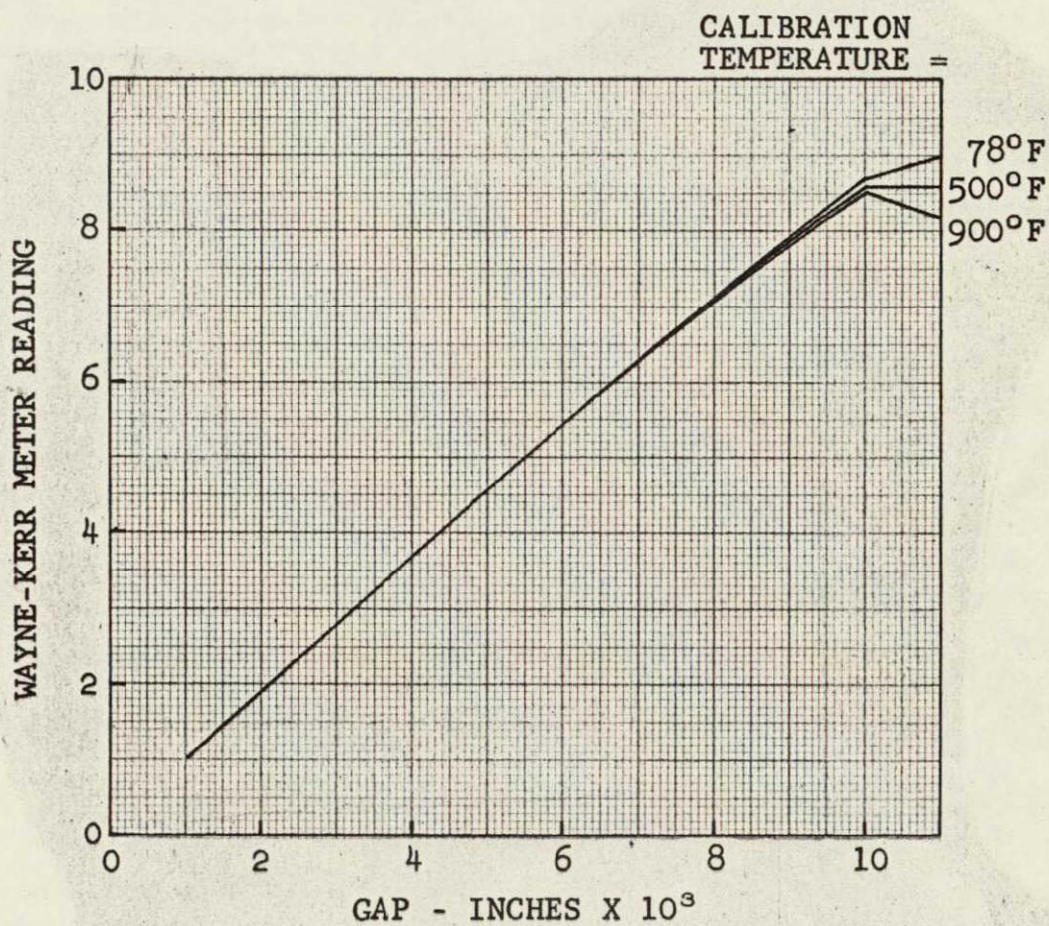
Tests on the completed gas-generator capacitance probes were performed and the results are summarized below:

- (a) Structural integrity is more than adequate for the operating range of temperatures within the gas generator.
- (b) Insulation resistance is fully compatible with the equipment to be used for readout purposes.
- (c) Calibration of the probes showed good linearity with acceptable temperature characteristics. Representative data is shown in Figure 37 for a flat-plate calibration.

A complete probe assembly is shown in Figure 38.



AIRESEARCH MANUFACTURING COMPANY OF ARIZONA
A DIVISION OF THE GARRETT CORPORATION
PHOENIX, ARIZONA



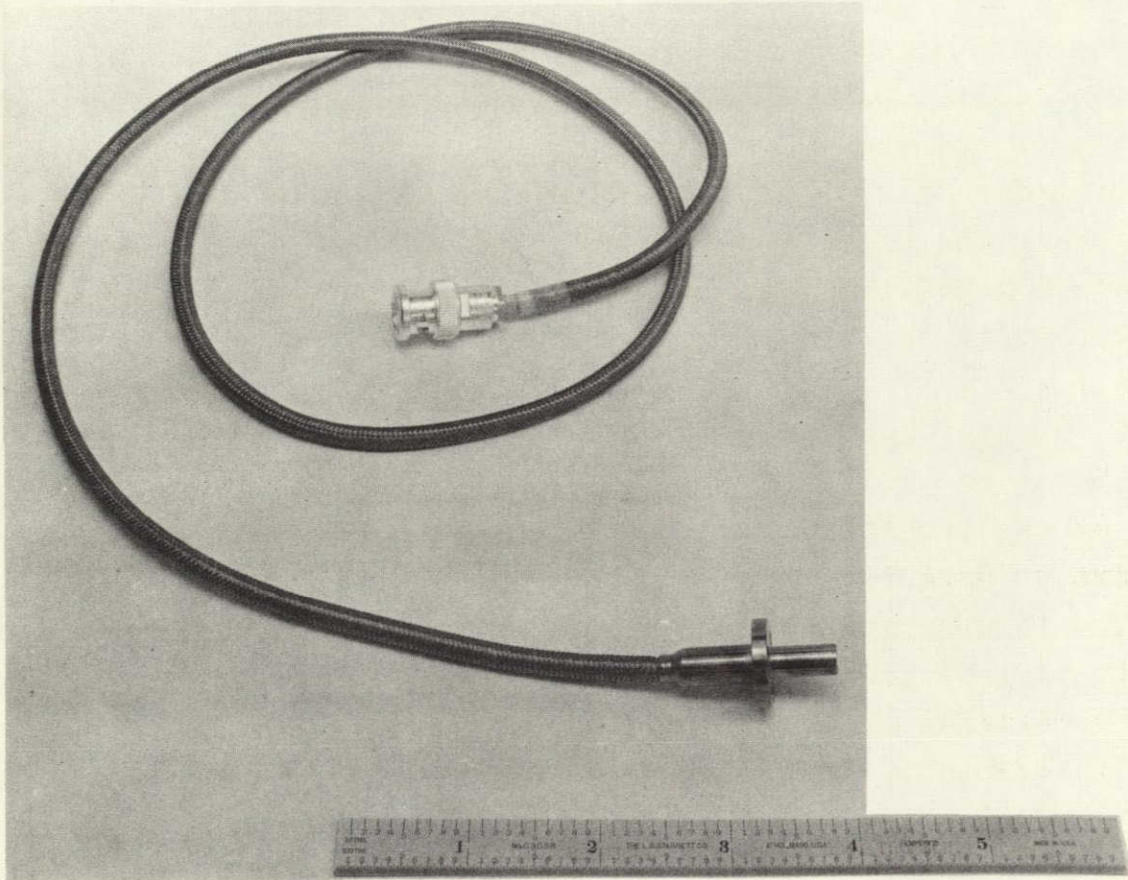
A32571

TEST RESULTS ON PROXIMITY PROBE CALIBRATIONS

FIGURE 37



AIRESEARCH MANUFACTURING COMPANY
A DIVISION OF THE GARRETT CORPORATION
PHOENIX, ARIZONA



SHAFT CAPACITANCE PROBE

FIGURE 38

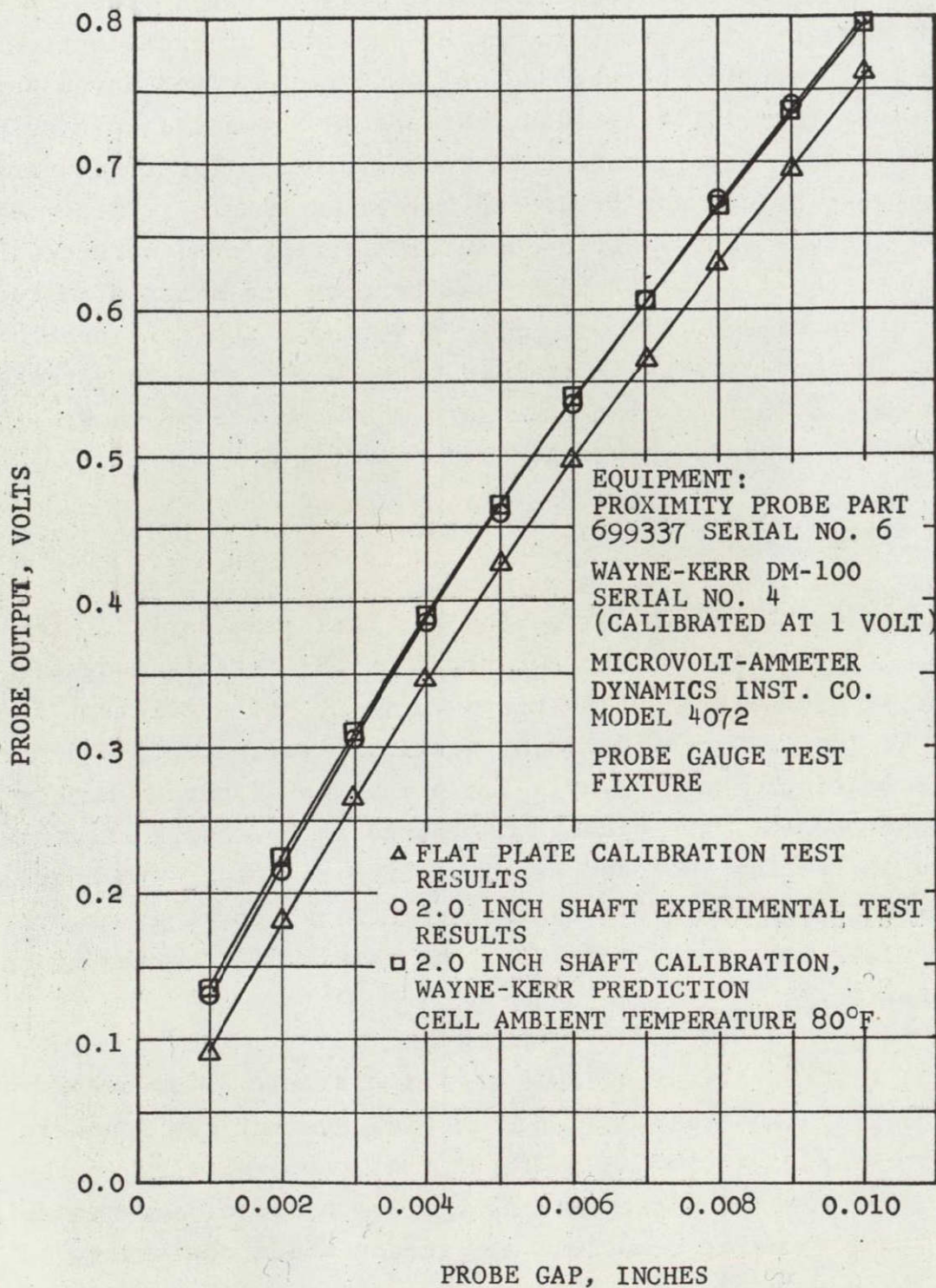


The test to determine the effect of a curved surface upon the calibration of a proximity probe showed that excellent correlation is obtainable between actual test results and Wayne-Kerr's predictions for similar probes. Figure 39 shows the test results obtained under normal ambient test conditions. The total variation between actual and predicted values is within $\pm 1/2$ percent of full scale. Wayne-Kerr's correction factor can therefore be applied to the flat-plate probe calibration for shaft diameters of the order of 2.00 inches. Figure 40 shows the test probe mounted in the calibration fixture, the curved surface representative of a 2.00-inch-diameter shaft, and the use of gauge-blocks to accurately set the probe gaps.

3.8.2 Strain Gauges

An experimental investigation was performed early in the program to establish whether or not four SR-4 strain gauges, mounted in quadrature on the plate surface, had sufficient sensitivity to indicate bearing load, bearing misalignment, and direction of bearing misalignment. Two prototype diaphragm mounts were instrumented, the diaphragms loaded in 5-pound increments, and the deflection recorded as a function of load. The angular stiffness of the diaphragm was determined by applying a moment at the plate center and measuring the angular deflection of the plate center.

The test results indicated that the strain gauge method for measuring bearing shoe load and angular misalignment has adequate sensitivity and provides a powerful tool for assisting in the aligning of the bearings, setting bearing preload, and measuring bearing loads during transient and steady-state operation.



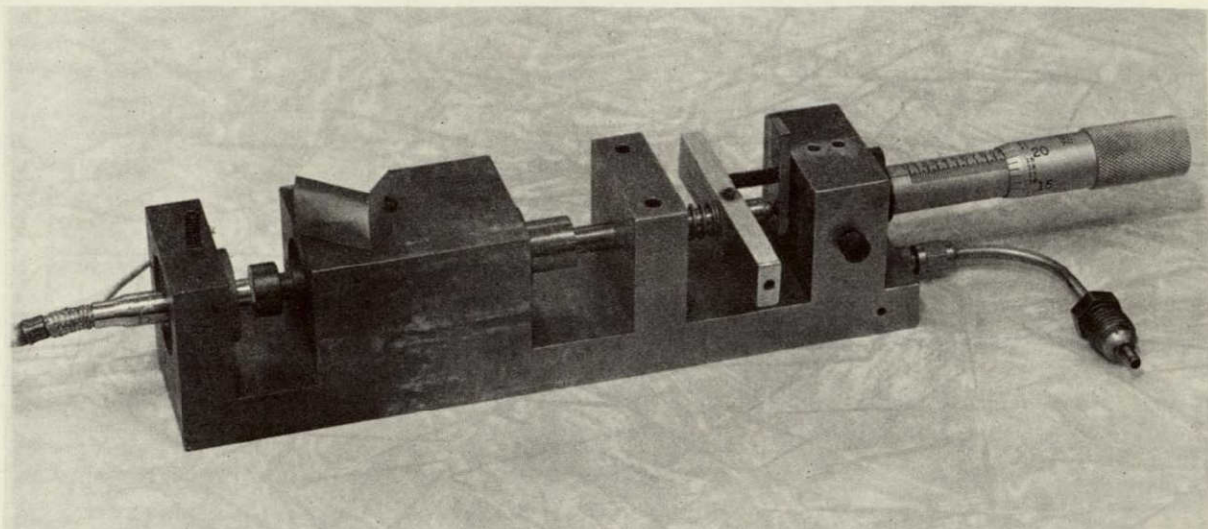
PREDICTED AND ACTUAL TEST RESULTS OF EFFECT OF CURVED SURFACE ON FLAT-PLATE CALIBRATION OF WAYNE-KERR PROXIMITY PROBE SYSTEM

FIGURE 39

A32573



AIRESEARCH MANUFACTURING COMPANY OF ARIZONA
A DIVISION OF THE GARRETT CORPORATION
PHOENIX, ARIZONA



PROXIMITY PROBE IN CALIBRATION FIXTURE

FIGURE 40



The utilization of strain gauges on the thrust bearing mount to determine thrust loading is similar to journal bearing use.

The three strain-gauge circuits comprise the conventional four-arm bridge network, with two opposite arms of the bridge being subjected to the readout system's excitation signal and the two remaining arms reading the output signal. A representation of the turbine-end shoe diaphragm circuit is shown in Figure 41.

The compressor-end shoe diaphragm circuit and thrust bearing load-sensing circuit are to be handled in a like manner. Calibrations of the strain gauges were provided with the final gas generator.

3.8.3 Thermocouples

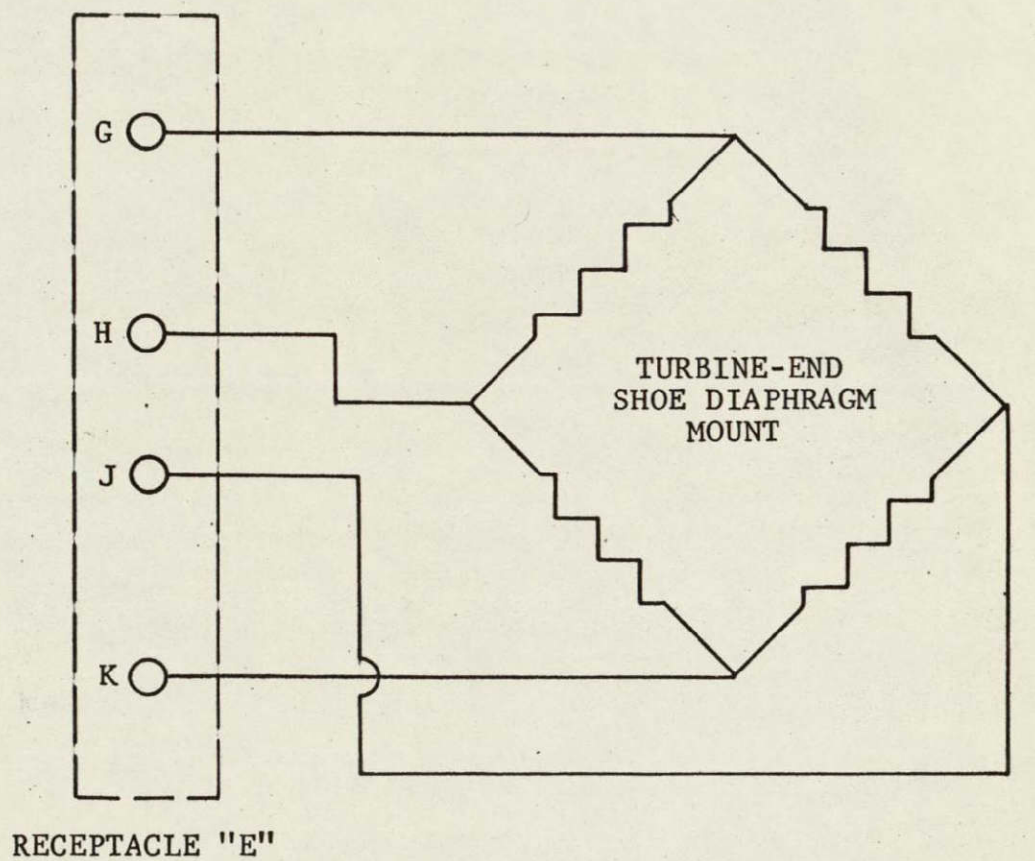
A total of 54 points were instrumented on the basis of the thermal analysis undertaken during the design phase of the program. However, 23 thermocouples were hooked up to the instrumentation receptacle plate for readout during the acceptance tests, as discussed in Section 5.0. The remaining 31 thermocouples were included since the incorporation of additional thermocouples would necessitate the complete disassembly of the gas generator at the NASA, thus resulting in possible delays during the subsequent testing program.

3.8.4 Additional Instrumentation

Figure 42 shows one of the magnetic speed pickups, three of which are incorporated into the gas generator. Figures 43, 44, 45, and 46 show the installation of the instrumentation prior to the acceptance test. The instrumentation leads are hooked to the instrumentation receptacle plate shown in Figure 47.



AIRESEARCH MANUFACTURING COMPANY OF ARIZONA
A DIVISION OF THE GARRETT CORPORATION
PHOENIX, ARIZONA



A32549

REPRESENTATIVE STRAIN-GAUGE CIRCUIT

FIGURE 41



AIRESEARCH MANUFACTURING COMPANY OF ARIZONA

A DIVISION OF THE GARRETT CORPORATION
PHOENIX, ARIZONA

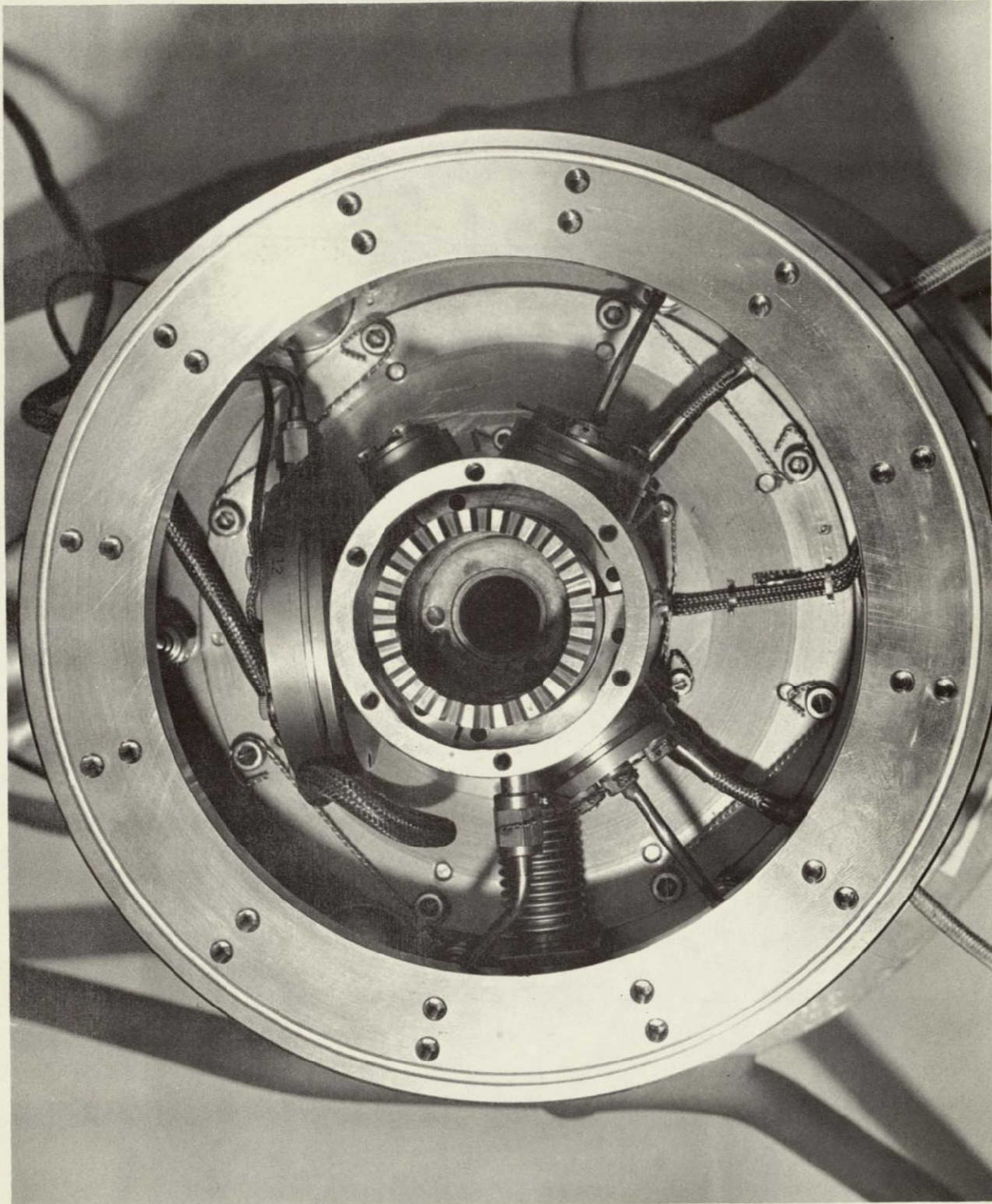


MAGNETIC SPEED PICKUP

FIGURE 42



AIRESEARCH MANUFACTURING COMPANY OF ARIZONA
A DIVISION OF THE GARRETT CORPORATION
PHOENIX, ARIZONA



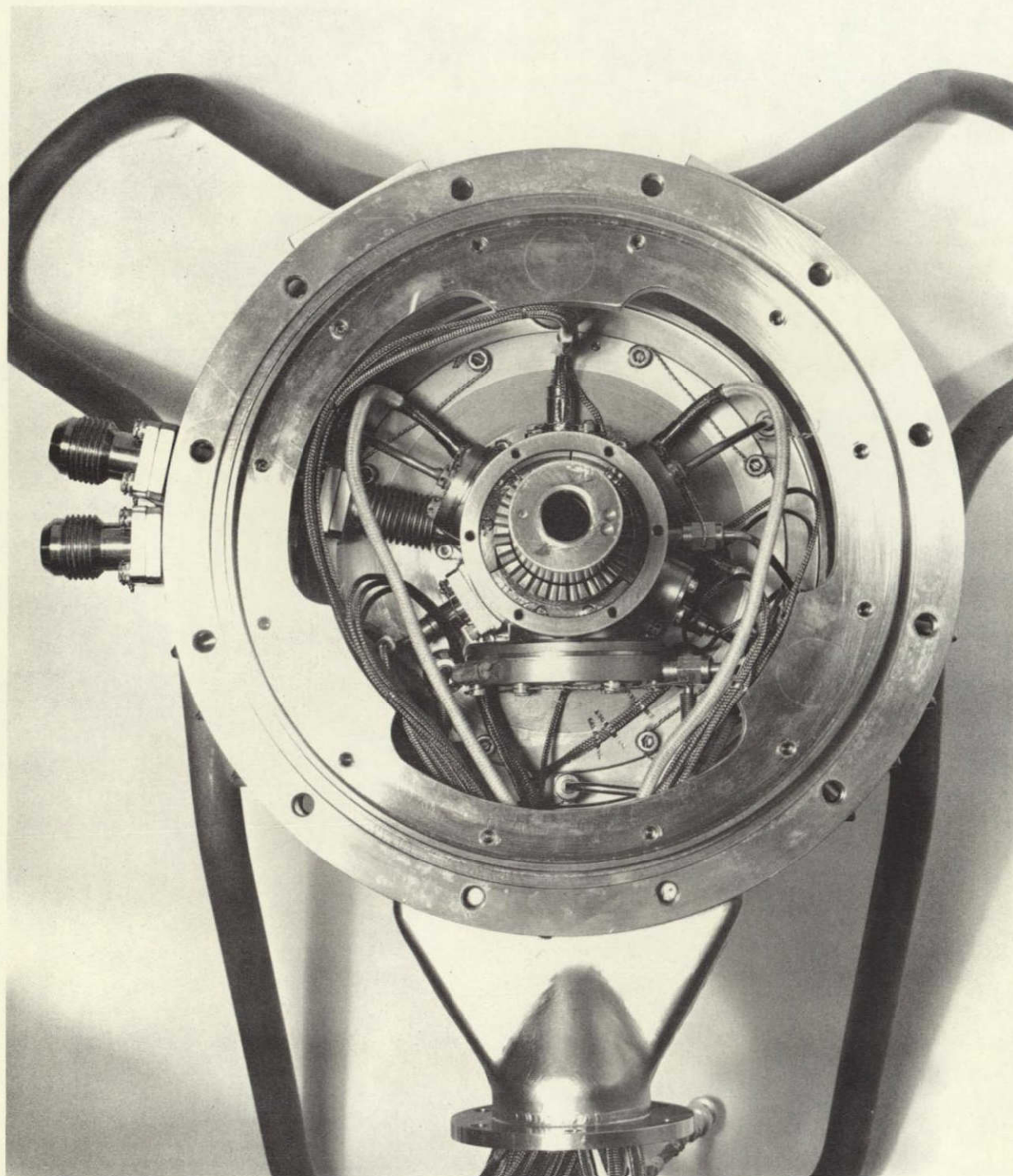
INSTRUMENTATION INSTALLATION, PARTIAL ASSEMBLY
NASA GAS GENERATOR

FIGURE 43



AIRESEARCH MANUFACTURING COMPANY OF ARIZONA

A DIVISION OF THE GARRETT CORPORATION
PHOENIX, ARIZONA

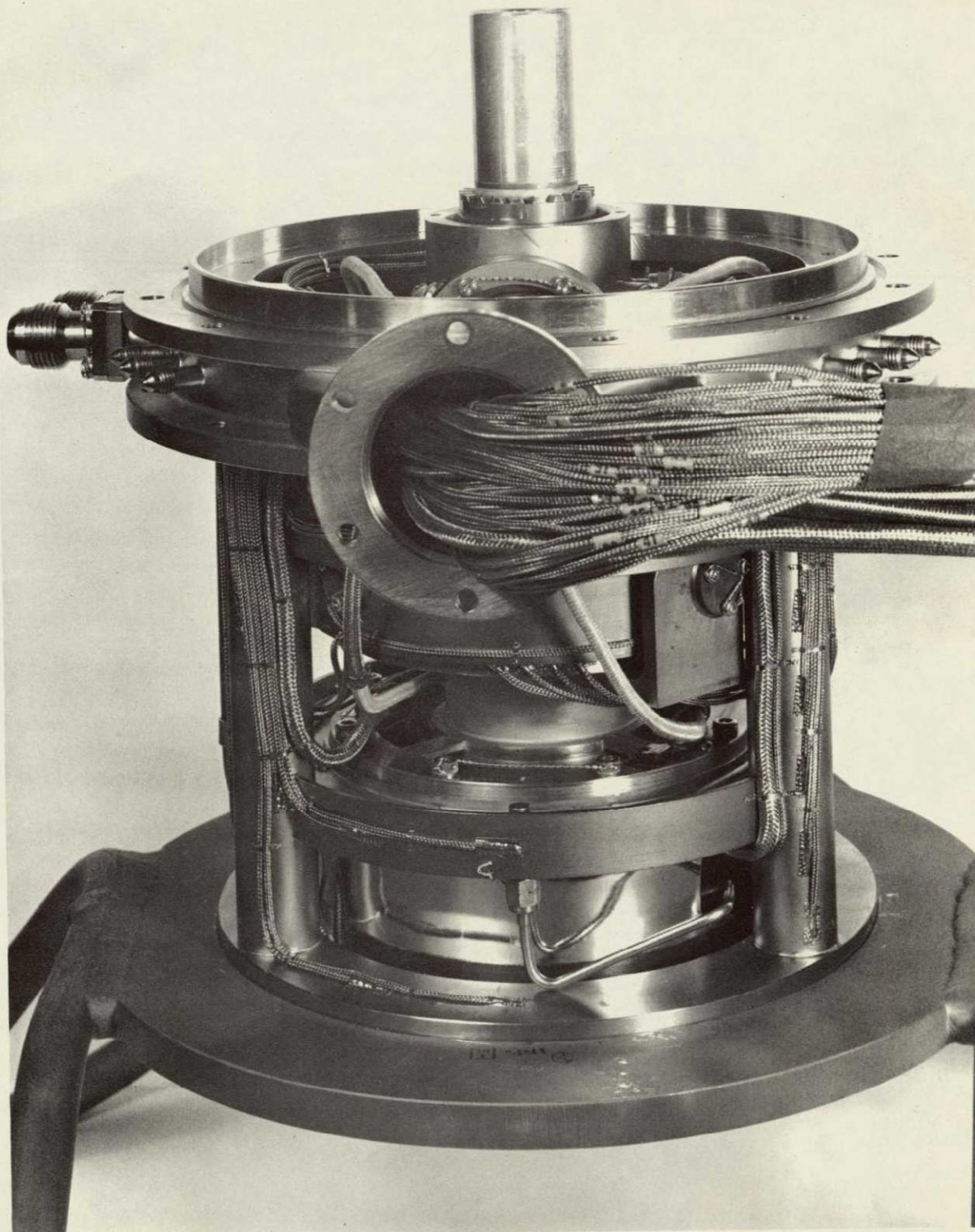


INSTRUMENTATION INSTALLATION, FINAL WIRING
NASA GAS GENERATOR

FIGURE 44



AIRESEARCH MANUFACTURING COMPANY OF ARIZONA
A DIVISION OF THE GARRETT CORPORATION
PHOENIX, ARIZONA



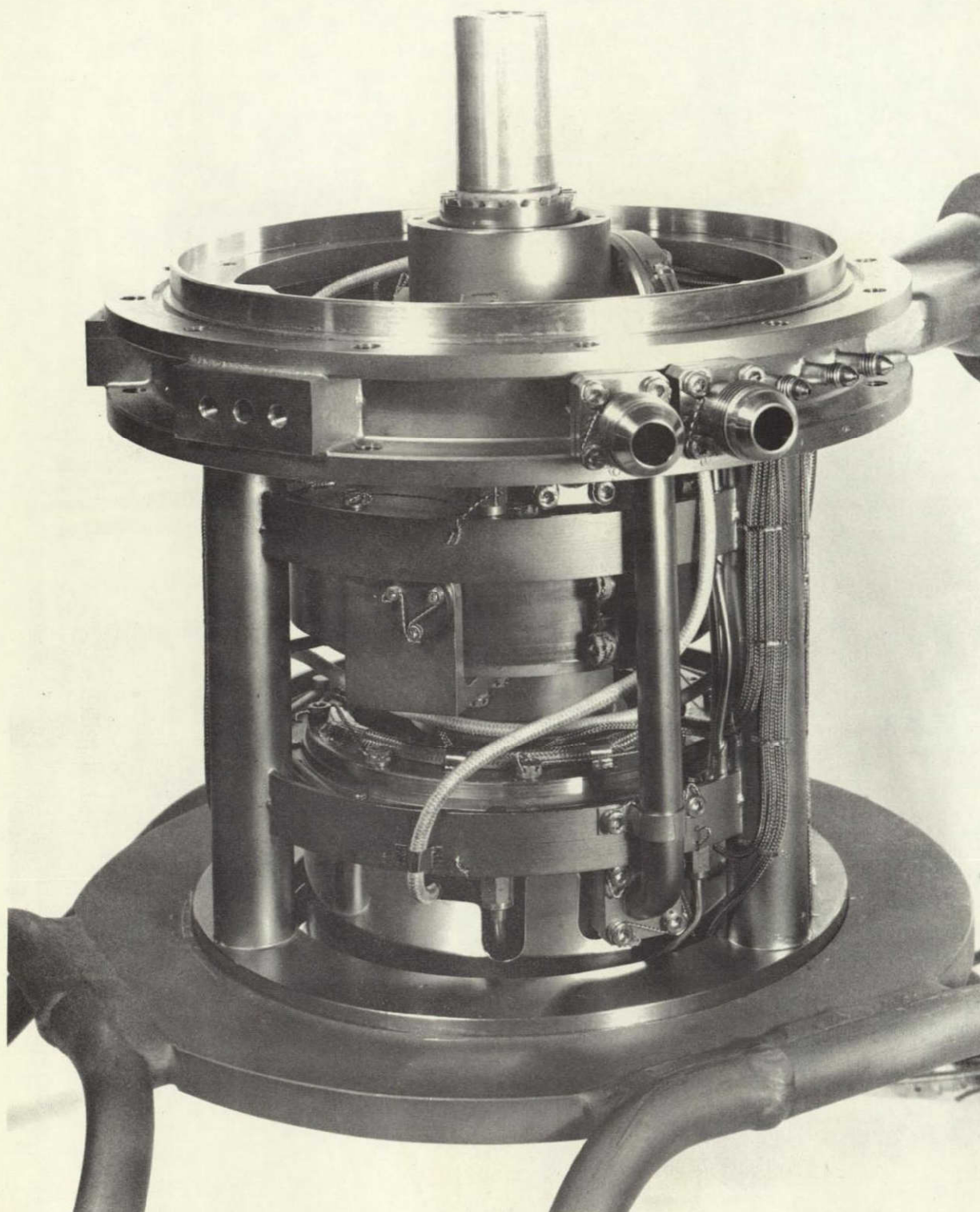
INSTRUMENTATION INSTALLATION, FINAL WIRING
NASA GAS GENERATOR

FIGURE 45



AIRESEARCH MANUFACTURING COMPANY OF ARIZONA

A DIVISION OF THE GARRETT CORPORATION
PHOENIX, ARIZONA



INSTRUMENTATION INSTALLATION, FINAL WIRING
NASA GAS GENERATOR

FIGURE 46

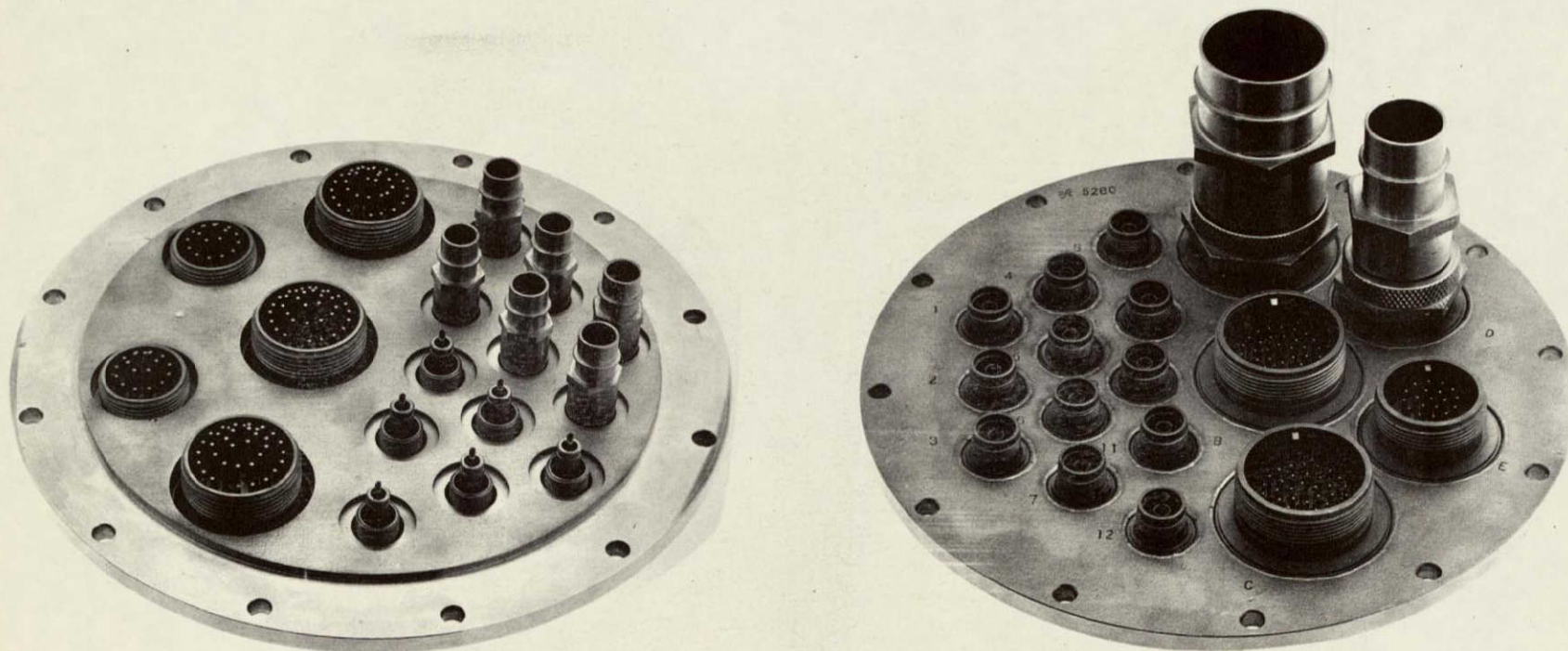


AIRESEARCH MANUFACTURING COMPANY

A DIVISION OF THE GARRETT CORPORATION

PHOENIX, ARIZONA

81



INSTRUMENTATION RECEPTACLE PLATE

FIGURE 47



In addition, the gas generator has been designed to accept gas bearing shoes with the following incorporated instrumentation:

- (a) Capacitance probes located in the bearing housing and situated such that they can monitor the bearing shoe leading edge of the first trailing shoe behind the resiliently mounted shoe on each bearing to determine pitch and roll characteristics of the bearings (four probes).
- (b) Capacitance probes built into the same shoes described above to determine the bearing shoe-to-shaft film thickness. The probe position will be on the pivot pitch axis as close to the pivot as possible (two probes).

This instrumentation would be utilized to determine gas bearing performance.



4.0 FINAL GAS GENERATOR CONFIGURATION

4.1 Gas Generator Description

Figure 48 presents a simplified cross section of the final gas generator. For purposes of clarity, the unit may be considered to consist of eight groups of parts. These are (1) the rotating assembly, (2) the compressor-journal bearing assembly, (3) the turbine-journal bearing assembly, (4) the gimbal and thrust bearing assembly, (5) the compressor scroll and diffuser assembly, (6) the turbine scroll and nozzle assembly, (7) the main frame assembly, and (8) the mount assembly. In the following paragraphs, each of the above subassemblies is described. A list of the individual parts is presented in Table 5.

4.1.1 Rotating Assembly

The rotating assembly consists of the shaft, the compressor wheel, and the turbine wheel. The compressor wheel and the turbine wheel are positioned on the shaft with curvic couplings. A tension bolt, drawn up by the self-locking shaft nut and washer, provides the axial force required to hold the assembly together as a rigid body. The compressor wheel spinner completes the rotating assembly and is attached to the end of the tension bolt with a self-locking screw.

4.1.2 Compressor/Turbine Journal Bearing Assembly

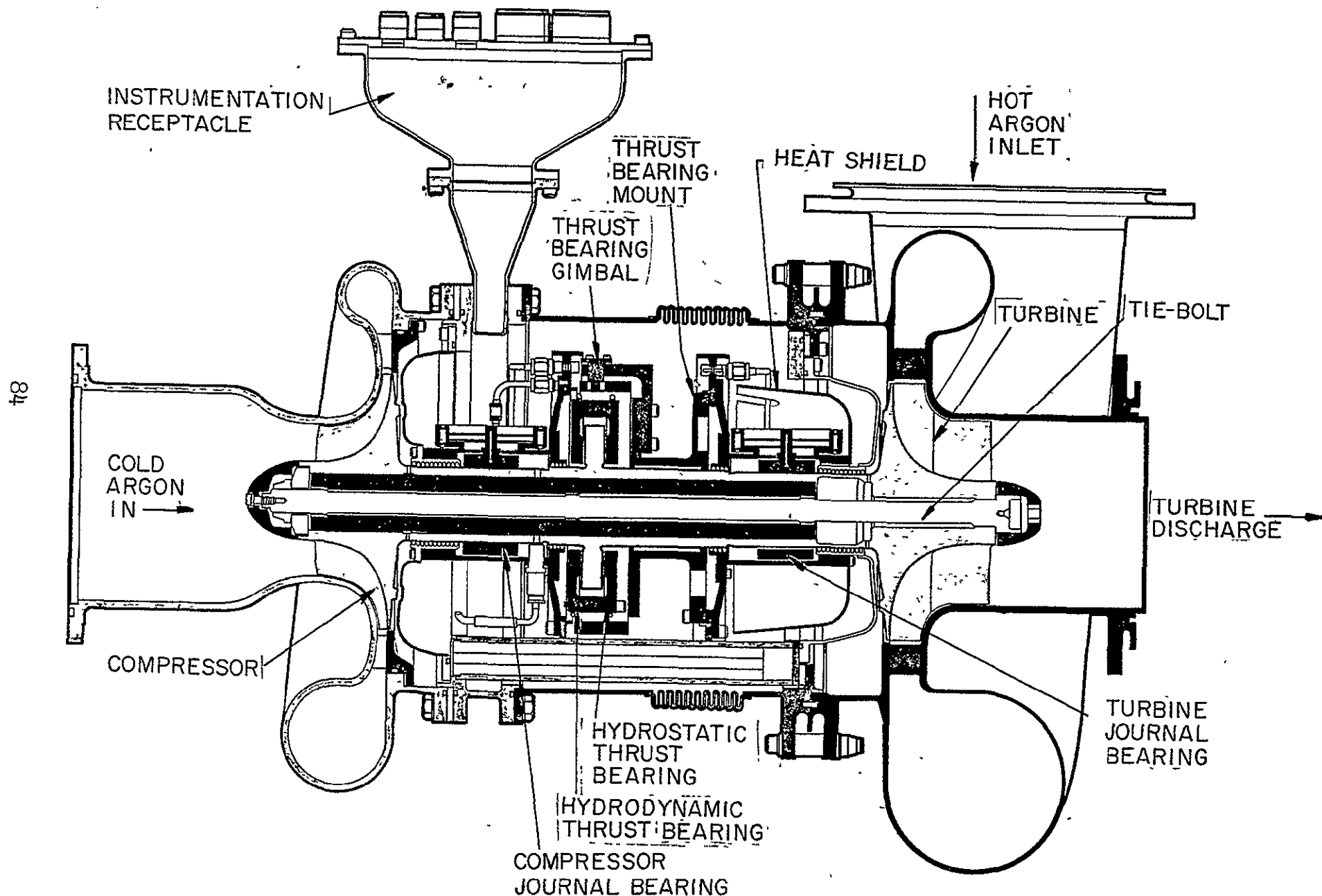
The compressor/turbine journal bearing assembly consists of the compressor/turbine journal bearing carrier, two shoe-and-fixed-mount assemblies, and one shoe-and-flexible-mount assembly.



AIRESEARCH MANUFACTURING COMPANY OF ARIZONA

A DIVISION OF THE GARRETT CORPORATION

PHOENIX, ARIZONA



NASA BRAYTON-CYCLE RADIAL GAS GENERATOR

FIGURE 118



AIRESEARCH MANUFACTURING COMPANY OF ARIZONA
A DIVISION OF THE GARRETT CORPORATION
PHOENIX, ARIZONA

TABLE 5
NASA GAS GENERATOR PARTS LIST

DRAWING NO.	TITLE	USED ON	NO. REQD.
699301-1	GAS GENERATOR ASSEMBLY		
699302-1	Scroll Assembly, Turbine	699301-1	1
699303-1	Wheel, Turbine	699301-1	1
699304-1	Impeller, Compressor	699301-1	1
699307-1	Seal, Static	699301-1	1
699307-2	Seal, Static	**	1
699307-3	Seal, Static	**	1
699308-1	Scroll Assembly, Compressor	699301-1	1
699309-1	Diffuser Assembly, Compressor	699301-1	1
699310-1	Shaft Assembly, Gas Generator	699301-1	1
699311-1	Bolt, Tension	699301-1	1
699312-1	Frame Assembly, Main	699301-1	1
699313-1	Carrier Assembly, Turbine Bearing	699312-1	1
699314-1	Carrier Assembly, Compressor Bearing	699312-1	1
699315-1	Mount Assembly, Bearing	699363-1	1
699317-1	Mount Assembly, Bearing	699364-1	1
699318-1	Tube Assembly, Thrust Bearing	699301-1	2
699320-1	Stator, Hydrostatic Thrust Bearing	699301-1	1
699321-1	Stator, Hydrodynamic Thrust Bearing	699301-1	1
699324-1	Seal Assembly, Turbine	699301-1	1
699325-1	Seal Assembly, Compressor	699301-1	1
699326-1	Shield, Turbine Radiation	699301-1	1
699327-1	Housing Assembly, Outer	699301-1	1
699328-1	Housing Assembly, Instrumentation	699301-1	1
699329-1	Plate Assembly, Connector	699301-1	1
699330-1	Shoe Assembly, Bearing	699363-1	1
699330-1	Shoe Assembly, Bearing	699364-1	1
699333-1	Tube Assembly, Metal Pressure Tap	699301-1	2
699334-1	Spinner, Impeller	699301-1	1
699335-1	Washer, Thrust	699301-1	1
699337-1	Probe, Capacitance (Shaft)	699301-1	4
699338-1	Probe, Capacitance (Thrust Bearing)	699301-1	2
699339-1	Pickup Assembly, Magnetic	699301-1	3
699340-1	Bellows, Duct	699313-1	1
699340-1	Bellows, Duct	699314-1	1
699342-1	Bellows, Outer Housing	699327-1	1
699343-1	Flange, Turbine Inlet	**	1
699344-1	Flange, Turbine Outlet	**	1
699345-1	Spacer, Sleeve	699301-1	12
699345-1	Spacer, Sleeve	**	16
699346-1	Shim, Bearing Mount	699301-1	2
699348-1	Shim, Bearing Mount	699301-1	4
699349-1	Shim, Turbine Scroll	699301-1	1
699354-1	Shim, Capacitance Probe	699301-1	4
699354-3	Shim, Capacitance Probe	699301-1	2
699355-1	Shim, Magnetic Pickup	699301-1	3
699356-1	Base, Mounting	699301-1	1
699357-1	Bracket Assembly, Support	699301-1	1
699358-1	Adapter, Fitting	699301-1	3
699359-1	Bracket Assembly, Mounting	699301-1	2
699360-1	Mount, Flexible Metal	699301-1	2
699363-1	Bearing, Matched Set	699301-1	4
699364-1	Bearing, Matched Set	699301-1	2
699365-1	Nut, Self Locking	699301-1	1
699366-1	Tube Assembly, Diaphragm Mount	699301-1	1
699368-1	Cover, Capacitance Probe Boss	699301-1	4
699376-1	Tube Assembly	699301-1	1
699410-1	Probe Assembly, Electrical Ground	699301-1	1
699411-1	Housing, Probe Support	699410-1	1
699412-1	Stem Assembly, Probe	699410-1	1
699413-1	Guide, Stem	699410-1	1
699424-1	Seal, Shaft	699301-1	2
699434-1	Pivot, Thrust Bearing Mount	699452-1	4
699435-1	Insert Pivot	699440-1	2
699436-1	Shim, Pivot	699452-1	4
699440-1	Ring Assembly, Thrust Bearing Mount	699452-1	1
699441-1	Spacer Assembly, Thrust Bearing	699452-1	1
699443-1	Bushing, Pivot	699441-1	2
699451-1	Mount Assembly, Thrust Bearing	699452-1	1
699452-1	Gimbal, Matched Set	699301-1	1



AIRESEARCH MANUFACTURING COMPANY OF ARIZONA
A DIVISION OF THE GARRETT CORPORATION
PHOENIX, ARIZONA

NASA GAS GENERATOR PARTS LIST (CONTD.)

DRAWING NO.	TITLE	USED ON	NO. REQD.
369745	Shim, Sealing Spacer	699301	1
369778	Flange, Compressor Outlet	**	1
369779	Flange, Compressor Inlet	**	1
369813	Seal, "O"-Ring	699301	3
362-506-9013	Gasket	699301	1
362-506-9014	Gasket	699301	1
362-506-9015	Gasket	699301	1
362-522-9006	Gasket	699301	2
362-542-9002	Gasket	699301	16
473-010-9001	Lubricant	699301	AR
525-578-9003	Nut	699301	12
655-601-9208	Screw	699301	1
S8171AP5	"O"-Ring	699301	1
S8171AP16	"O"-Ring	**	1
S8171AP30	"O"-Ring	**	1
S8171AP30	"O"-Ring	699301	1
S8171AP37	"O"-Ring	699301	1
S8152BG46-0-240	Pin	699312-1	4
S8152BC46-0-240	Pin	699451-1	4

COMMERCIAL PARTS

AN4CH4A	Bolt	699301	4
AN5CH4A	Bolt	699301	12
AN5CH5A	Bolt	699301	12
AN929-10S	Cap Assembly	699301	1
AN960C416L	Washer	699301	16
AN960C516	Washer	699301	14
AN906C516L	Washer	699301	12
MS9068-015	"O"-Ring	699301	2
MS20995C32	Lockwire	699301	AR
MS21045C4	Nut	699301	8
MS21281-25	Screw	699301	12
MS21288-06	Screw	699301	12
MS21288-07	Screw	699301	12
MS24585C51	Spring	699410-1	1
MS24673-1	Screw	699301	24
MS24673-1	Screw	699312-1	16
MS24673-2	Screw	699301	18
MS24673-2	Screw	699451-1	8
MS24673-3	Screw	699301	8
MS24673-9	Screw	699301	2
MS24673-9	Screw	699440-1	4
MS24673-11	Screw	699301	4
MS24673-12	Screw	699301	8
MS24673-22	Screw	699301	2
MS24674-1	Screw	699301	18
MS24674-2	Screw	699301	6
MS24674-3	Screw	699301	18
MS24674-6	Screw	699312-1	4
MS24674-7	Screw	699301	32
MS24674-22	Screw	699301	2
MS24677-1	Screw	699301	18
MS24677-2	Screw	699301	12
MS24677-2	Screw	699452-1	12
MS24678-15	Screw	699301	8
MS29513-004	"O"-Ring	699410-1	1

**Shipped with unit



A shim provides radial adjustment capability in each fixed-mount assembly. The axial center line of the rotating assembly is positioned by adjusting the two fixed mounts relative to the bearing carrier. A shim provides for bearing preload adjustment through the flexibly mounted shoe. Hydrostatic bearing gas is supplied to each fixed-mount bearing shoe by supply lines incorporated into the fixed mount and to the flexibly mounted bearing shoe by a tube assembly.

Two orthogonal capacitance probes, for monitoring shaft position, are provided at each bearing and adjusted by shims.

Three speed pickups are provided on the compressor journal bearing carrier and adjusted by shims.

4.1.3 Gimbal and Thrust Bearing Assembly

A gimbal assembly is provided to give the two thrust bearing stators alignment capability. Hydrostatic bearing gas is supplied individually to each thrust stator through tube assemblies.

A film-thickness capacitance probe and shim on each stator allow the bearing film thicknesses to be monitored during operation.

Attachment of the thrust bearing and gimbal assembly to the main frame is accomplished with the thrust bearing mount assembly into which are incorporated strain gauges for determining the thrust loads.



4.1.4 Compressor Scroll and Diffuser Assembly

The diffuser is attached by screws to the compressor scroll and sealed by an elastomer O-ring. The compressor discharge cooling-flow bleed fitting is located on the scroll discharge from which the cooling flow is ducted to the cooling inlet lines on the unit.

Axial face clearance between the wheel and the scroll is determined by sizing a shim located at the scroll mounting flange. The axial clearance was set at 0.012 inch during the design of the gas generator. From an aerodynamic consideration, a zero clearance would be optimum; however, a clearance of 0.002 inch per inch of wheel diameter can be utilized without serious performance penalty. From a mechanical consideration, it is advantageous to maintain large clearances so that, with rotor radial and axial displacements due to tolerance stackup, thermal growth, and flexible bearing displacements, the rotor does not rub the shroud.

For the acceptance test gas generator, the axial shroud clearance (for the turbine and the compressor) was increased over the design requirements to allow for test experience to be gained on the unit prior to extracting final efficiency figures.

4.1.5 Turbine Scroll and Nozzle Assembly

The turbine scroll and nozzle assembly is an integral one-piece assembly fabricated from an investment casting and several sheet-metal stampings. The axial face clearance between the turbine wheel and the scroll assembly is determined by sizing a shim located adjacent to the scroll mounting flange.



The axial clearance was set at 0.014 inch during the design. However, this clearance was increased for the acceptance test, as discussed above.

4.1.6 Main Frame Assembly

The main frame assembly serves two functions. . First and foremost, it provides a framework to which all major components or subassemblies are attached. This assures extreme accuracy in aligning those parts that have critical running clearances, such as the turbine scroll, the compressor scroll, the turbine labyrinth seal, and the compressor labyrinth seal.

Secondly, it contains all of the hydrostatic bearing gas supply lines, the cooling flow ducts and static pressure tap plumbing and provisions for routing and leading out all the electrical instrumentation lines.

4.1.7 Mount Assembly

An assembly is provided for mounting the unit in a vertical position with the turbine end down. The mount assembly consists of a base, a bracket, two flexible mounts, and two flexible mount brackets. The flexibility in the brackets and the two flexible mounts is intended to accommodate thermal expansions that occur during elevated-temperature operation of the unit.



4.2 Component Configurations and Fabrication

The compressor and turbine aerodynamic components utilized in the final gas generator are described in detail in NASA Reports CR-54368 and CR-54367, respectively. The journal gas bearing configuration is described in Section 3.4.4 of this report, and the thrust bearing configuration is described in Section 3.5.1. Figure 49 shows a simplified cross-section of the gas generator with the materials of more pertinent parts noted.

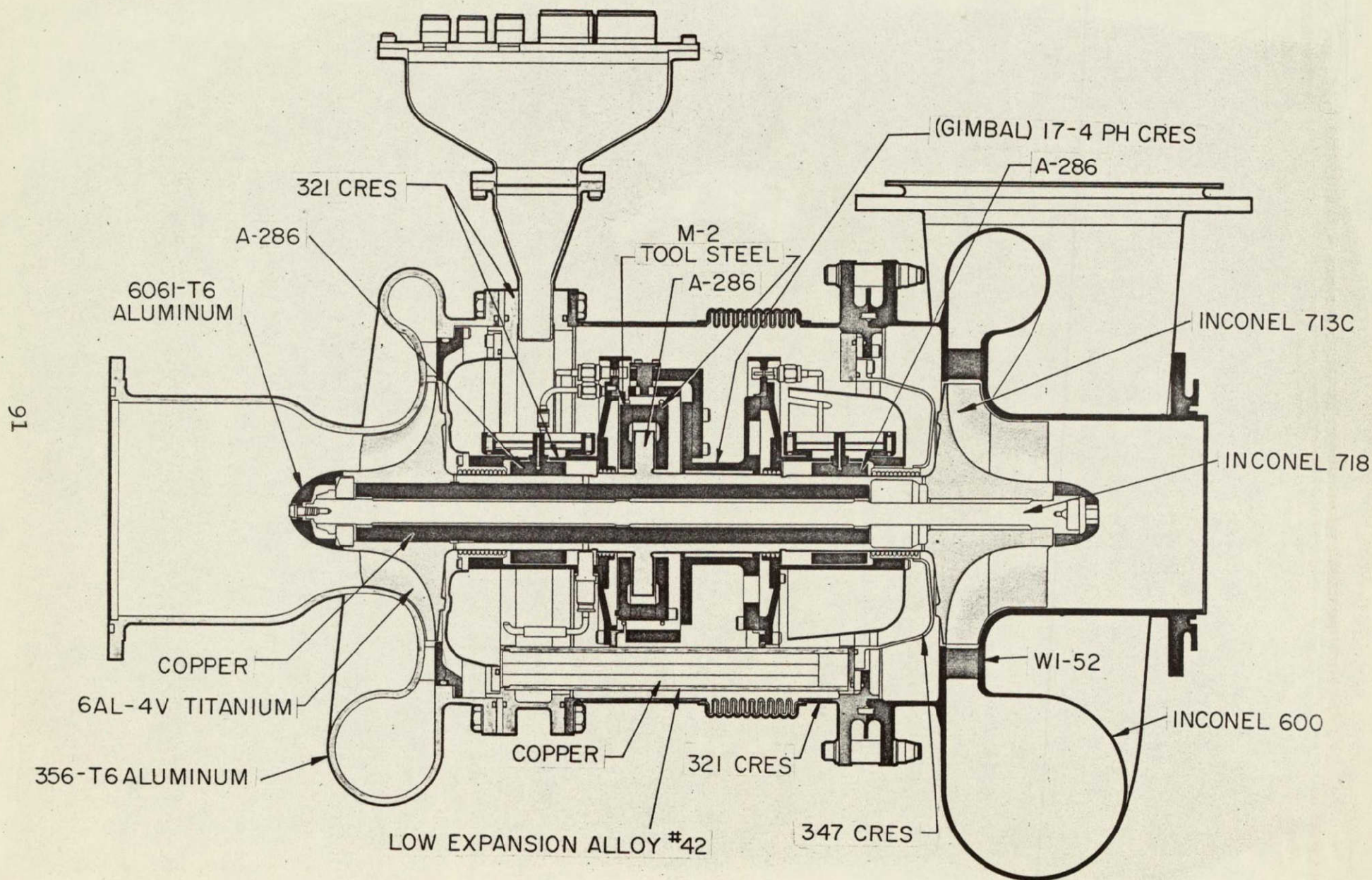
Fabrication of the gas generator components was initiated as the design of the components was finalized. Figures 50 and 51 show the major components after fabrication prior to the acceptance test. Details of some of the major components are shown in Figures 52 through 64, and Figure 65 shows the assembled gas generator.

4.3 Inspection

Inspection of the components of the delivery gas generators and final assembly were in accordance with the quality-assurance program established at the start of the program. Figures 66 and 67 show the Critical Parts Inspection and Serialization record for the turbine and compressor wheels of both delivery units. Figure 68 shows both sides of the Assembly Inspection and Laboratory Traveler for the gas generator, Serial No. P-A, assembled and operated for an acceptance test. The second gas generator, Serial No. P-B, was shipped as "component parts" at the request of the NASA; therefore, an Assembly Inspection and Laboratory Traveler is not available for this unit.



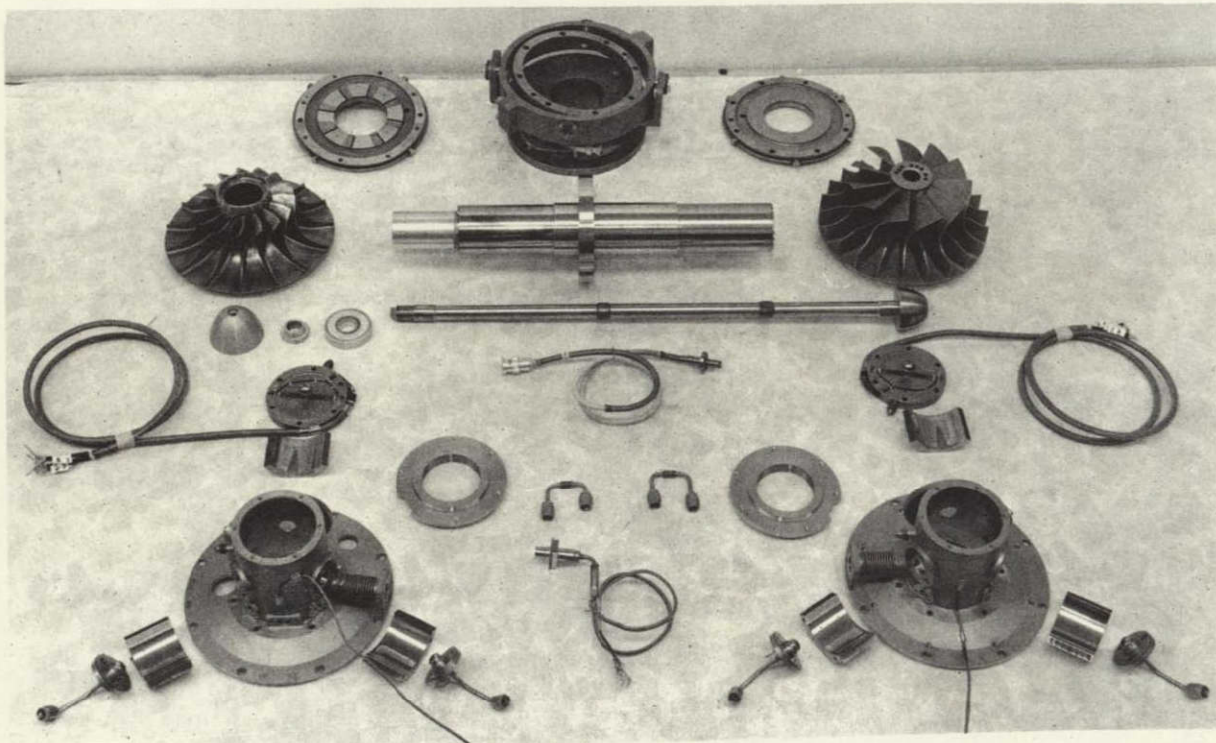
AIRESEARCH MANUFACTURING COMPANY OF ARIZONA
A DIVISION OF THE GARRETT CORPORATION
PHOENIX, ARIZONA



NASA BRAYTON-CYCLE RADIAL GAS GENERATOR
WITH MATERIALS OF PERTINENT PARTS NOTED
FIGURE 49



AIRESARCH MANUFACTURING COMPANY OF ARIZONA
A DIVISION OF THE GARRETT CORPORATION
PHOENIX, ARIZONA

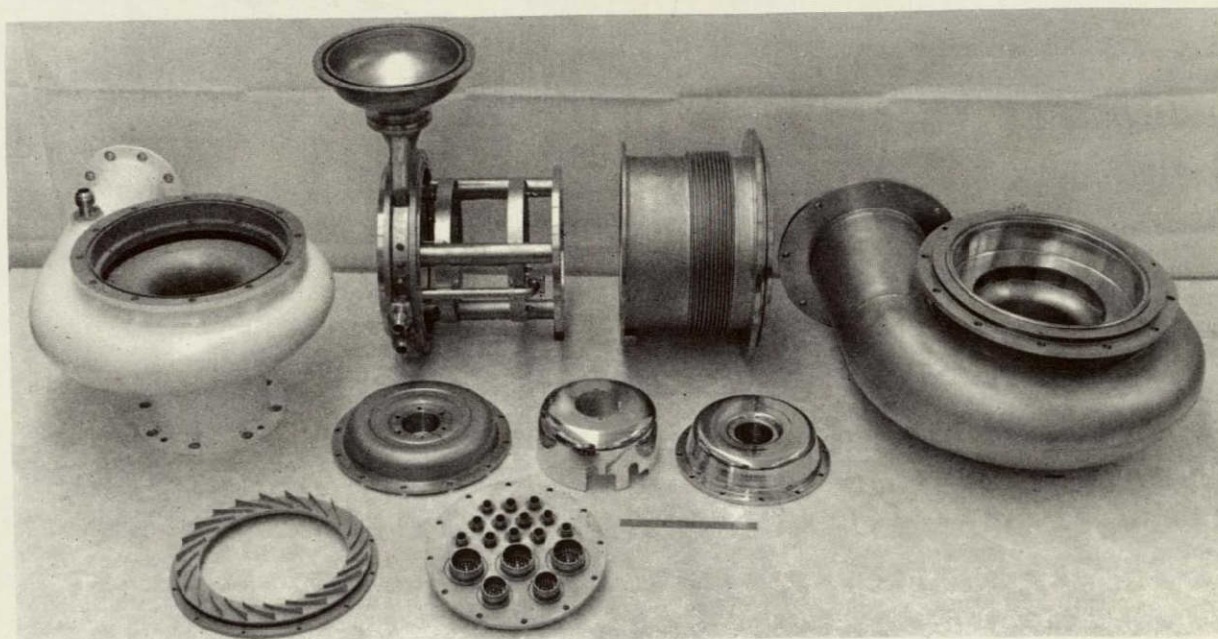


NASA GAS GENERATOR PART 699300
COMPONENT DISASSEMBLY

FIGURE 50



AIRESEARCH MANUFACTURING COMPANY OF ARIZONA
A DIVISION OF THE GARRETT CORPORATION
PHOENIX, ARIZONA



NASA GAS GENERATOR PART 699300
COMPONENT DISASSEMBLY

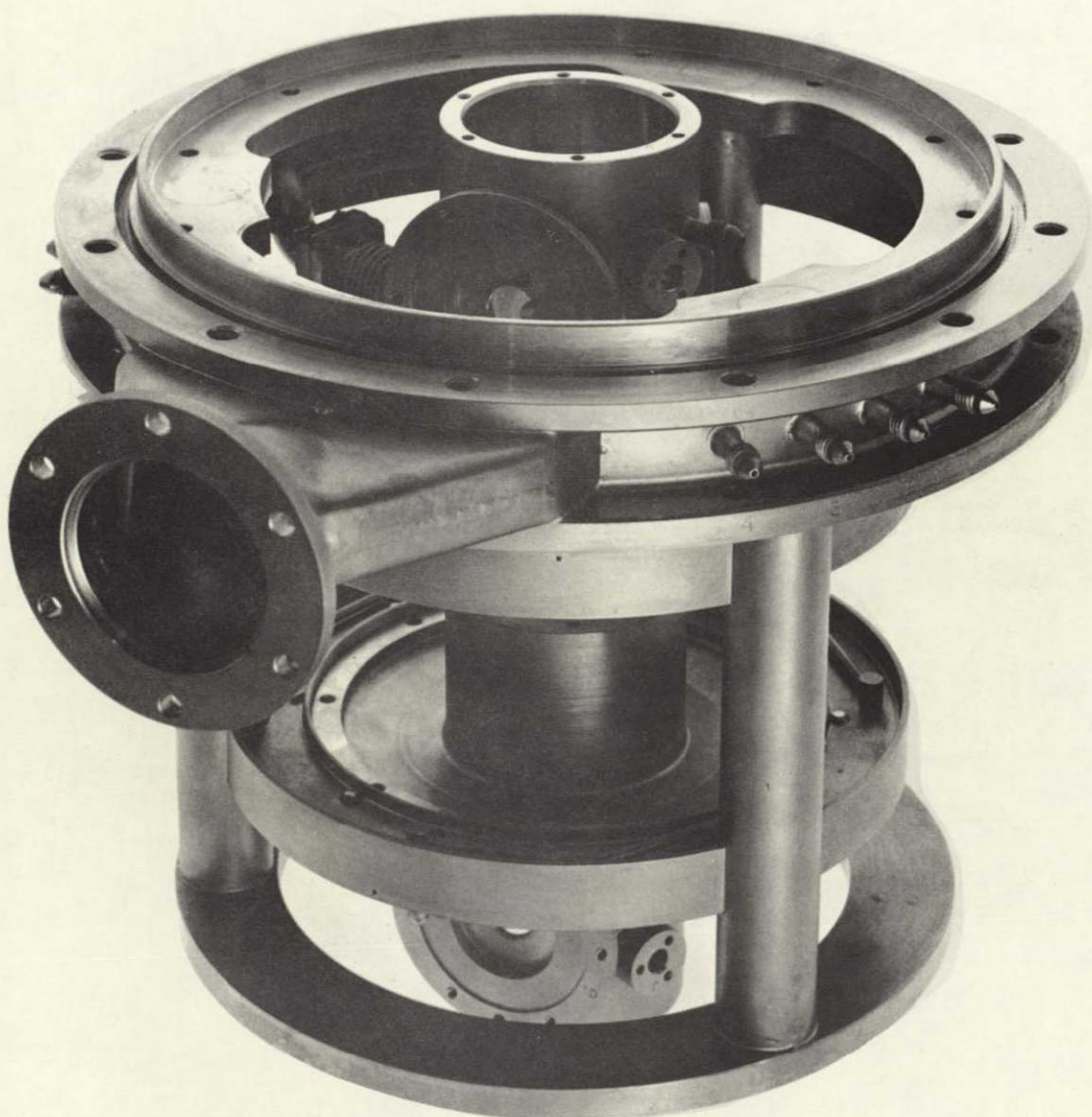
FIGURE 51



AIRESEARCH MANUFACTURING COMPANY OF ARIZONA

A DIVISION OF THE GARRETT CORPORATION

PHOENIX, ARIZONA

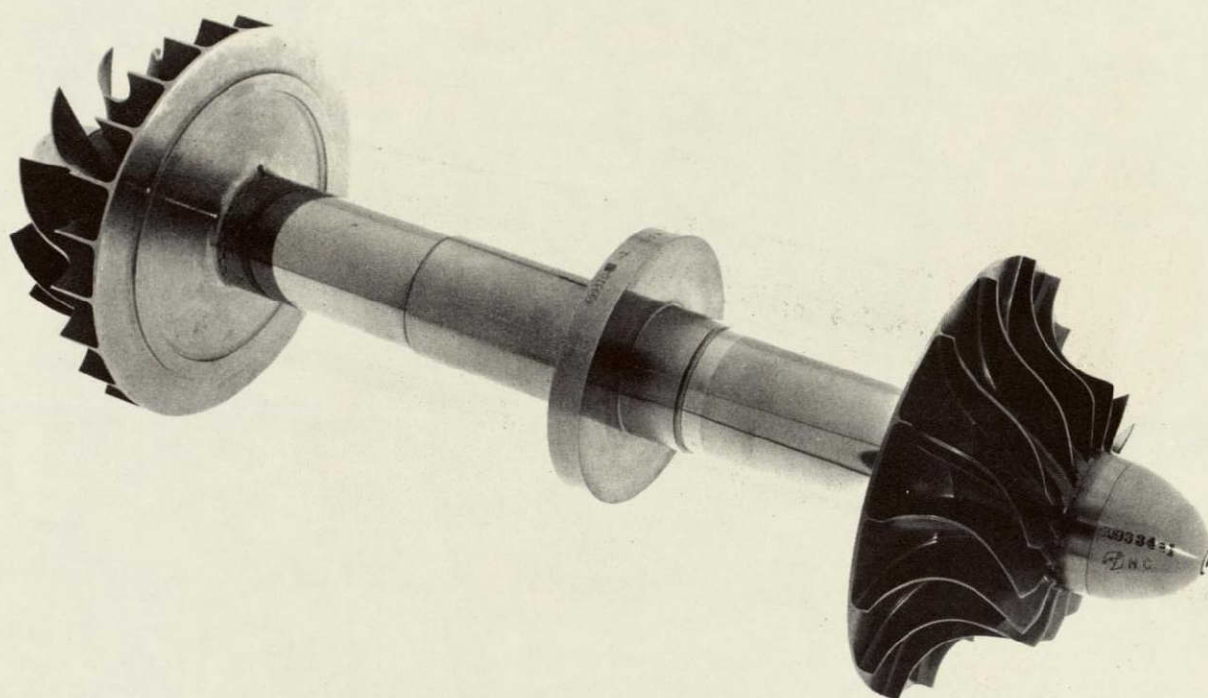


BEARING CARRIER AND FRAME ASSEMBLY

FIGURE 52



AIRESEARCH MANUFACTURING COMPANY OF ARIZONA
A DIVISION OF THE GARRETT CORPORATION
PHOENIX, ARIZONA

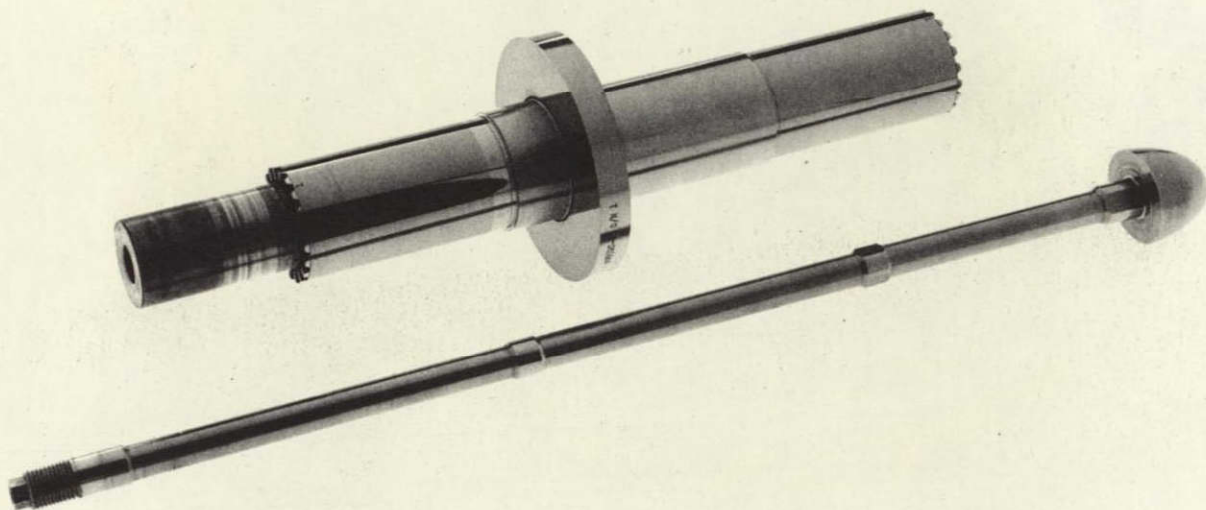


ROTATING COMPONENT GROUP ASSEMBLY

FIGURE 53



AIRESEARCH MANUFACTURING COMPANY OF ARIZONA
A DIVISION OF THE GARRETT CORPORATION
PHOENIX, ARIZONA

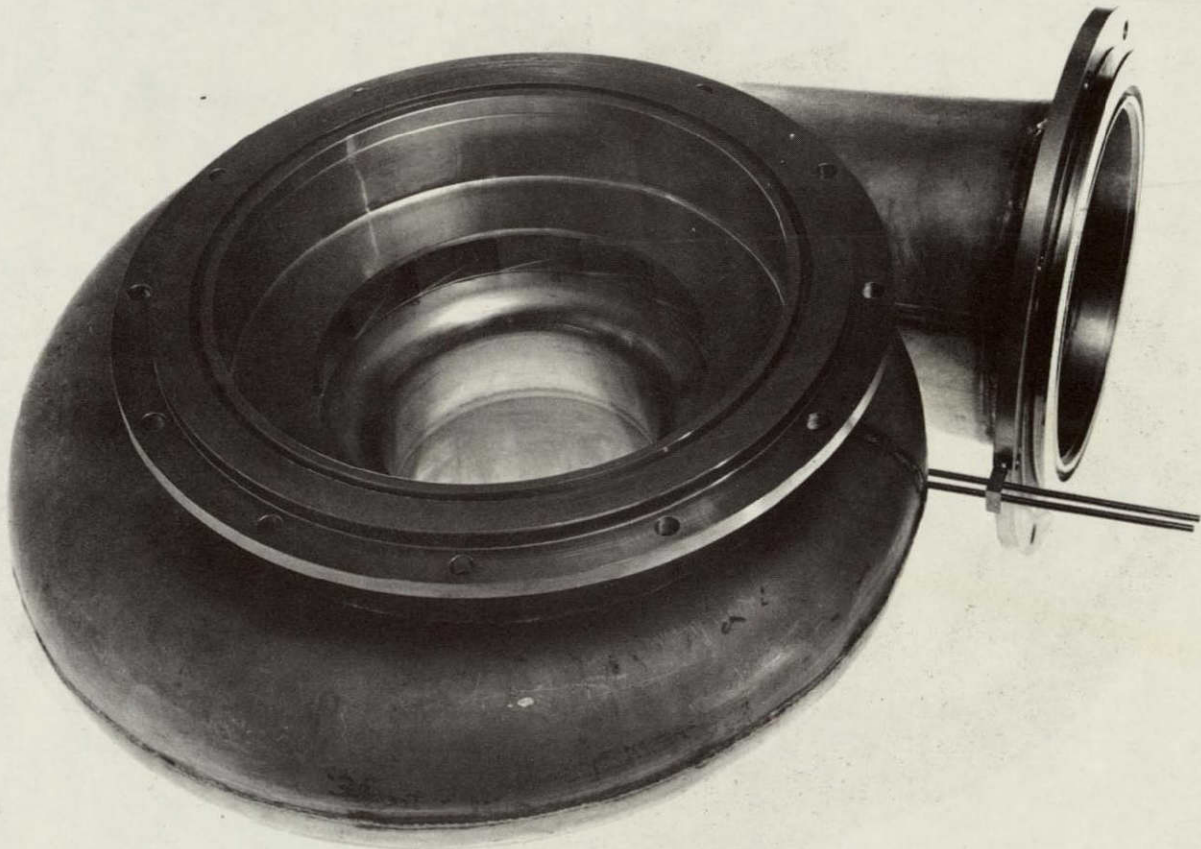


SHAFT ASSEMBLY AND TENSION BOLT

FIGURE 54



AIRESEARCH MANUFACTURING COMPANY OF ARIZONA
A DIVISION OF THE GARRETT CORPORATION
PHOENIX, ARIZONA

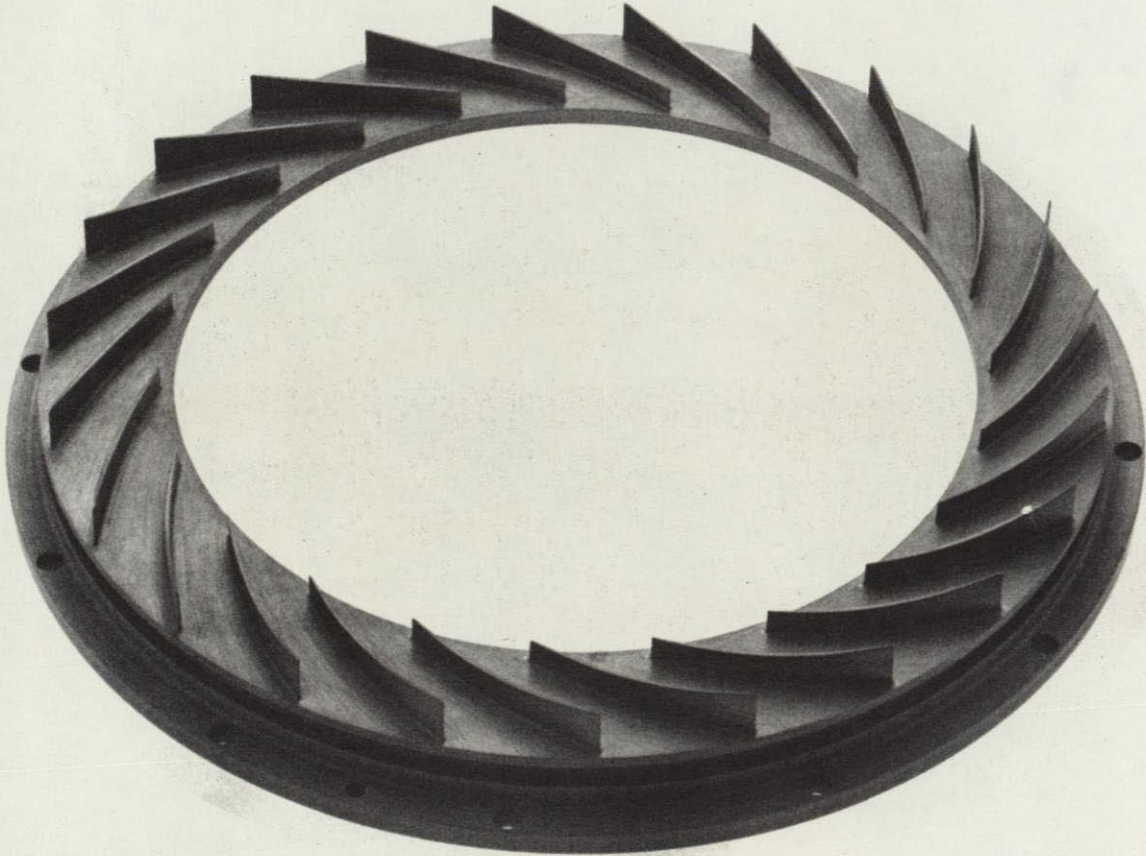


TURBINE SCROLL AND NOZZLE ASSEMBLY

FIGURE 55



AIRESEARCH MANUFACTURING COMPANY OF ARIZONA
A DIVISION OF THE GARRETT CORPORATION
PHOENIX, ARIZONA



COMPRESSOR DIFFUSER ASSEMBLY

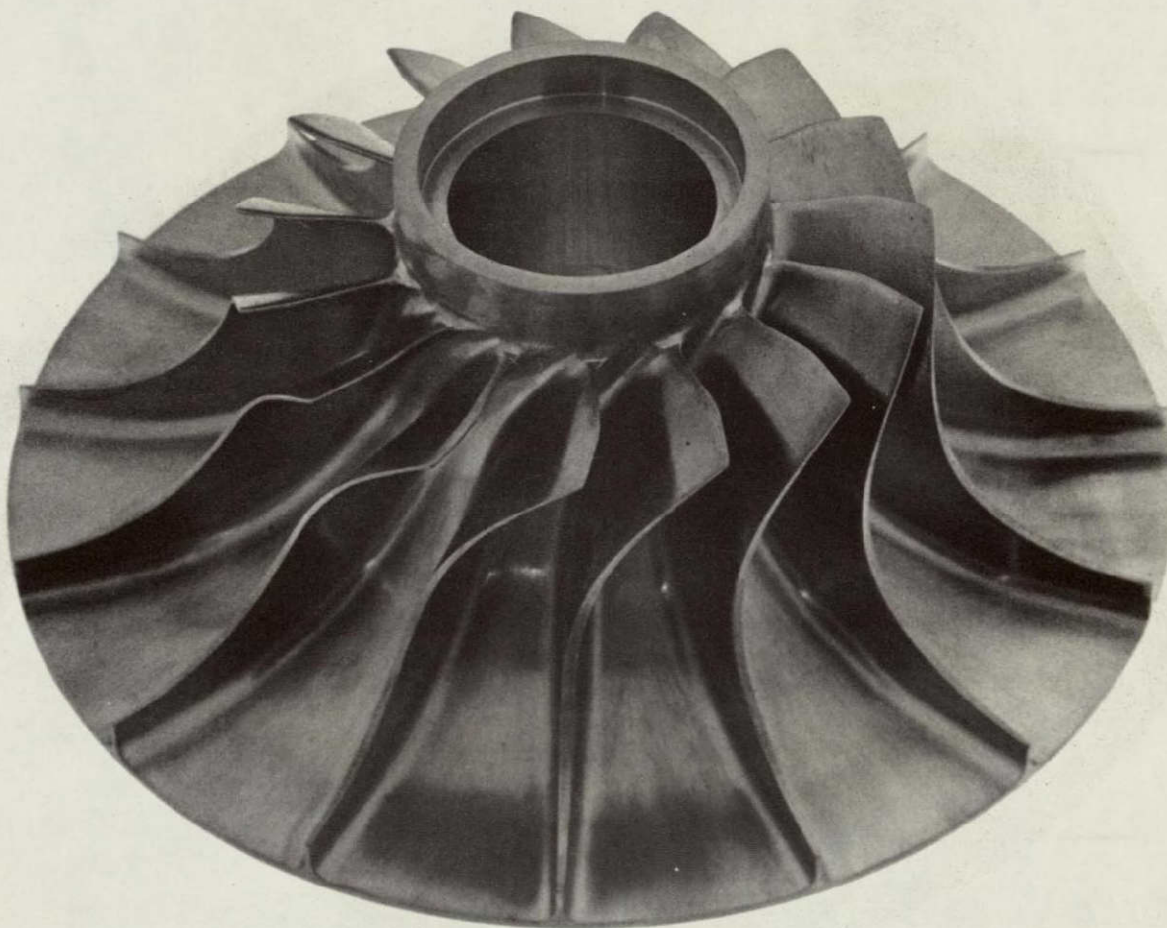
FIGURE 56



AIRESEARCH MANUFACTURING COMPANY OF ARIZONA

A DIVISION OF THE GARRETT CORPORATION

PHOENIX, ARIZONA



COMPRESSOR IMPELLER

FIGURE 57



AIRESEARCH MANUFACTURING COMPANY OF ARIZONA
A DIVISION OF THE GARRETT CORPORATION
PHOENIX, ARIZONA

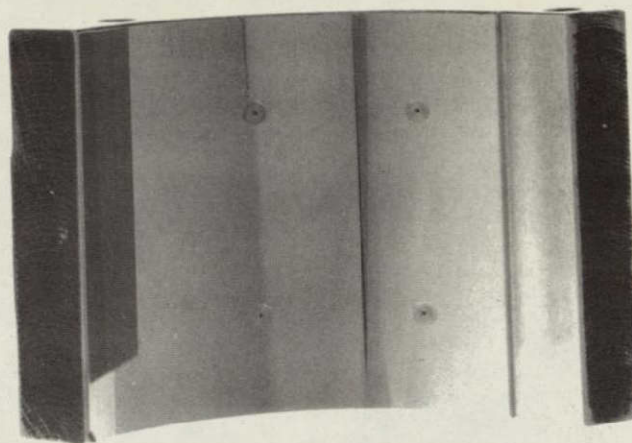
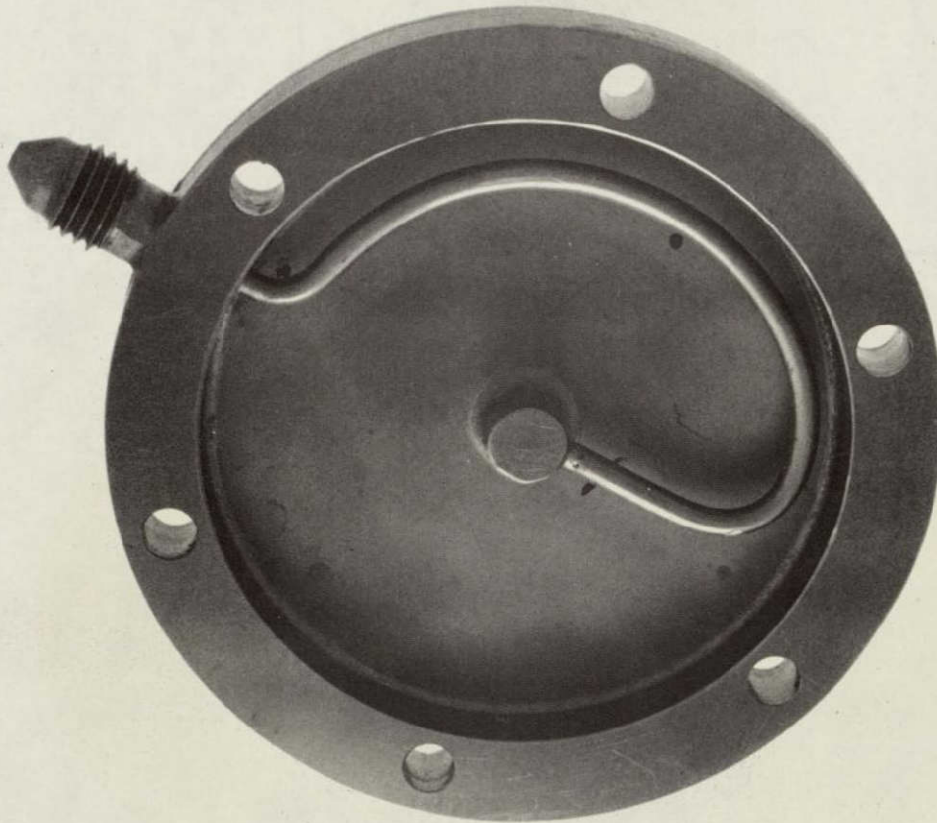


COMPRESSOR SCROLL ASSEMBLY

FIGURE 58



AIRESEARCH MANUFACTURING COMPANY OF ARIZONA
A DIVISION OF THE GARRETT CORPORATION
PHOENIX, ARIZONA

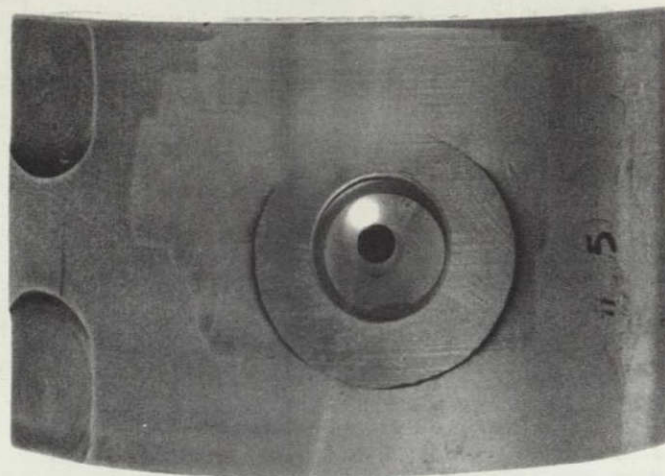
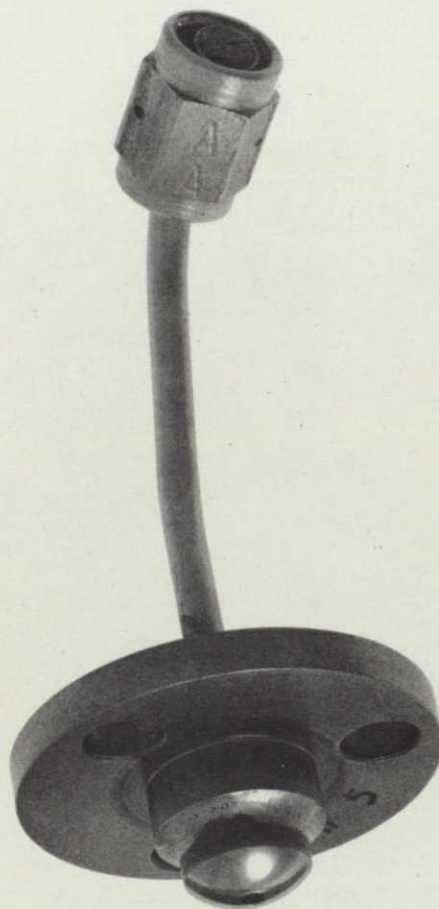


SHOE AND DIAPHRAGM ASSEMBLY, MATCHED SET

FIGURE 59



AIRESEARCH MANUFACTURING COMPANY OF ARIZONA
A DIVISION OF THE GARRETT CORPORATION
PHOENIX, ARIZONA



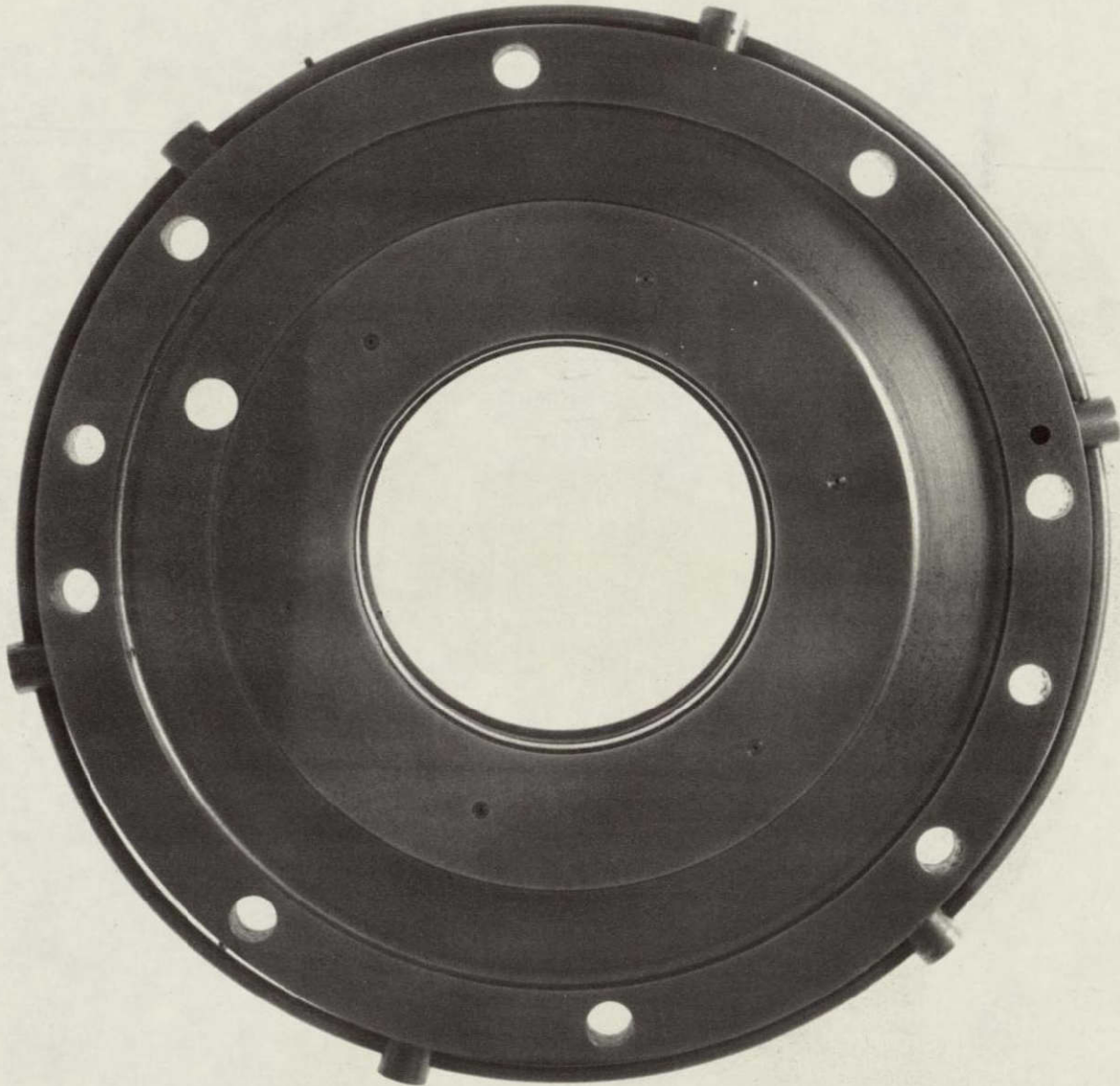
SHOE AND SOLID MOUNT, MATCHED SET

FIGURE 60



AIRESEARCH MANUFACTURING COMPANY OF ARIZONA

A DIVISION OF THE GARRETT CORPORATION
PHOENIX, ARIZONA

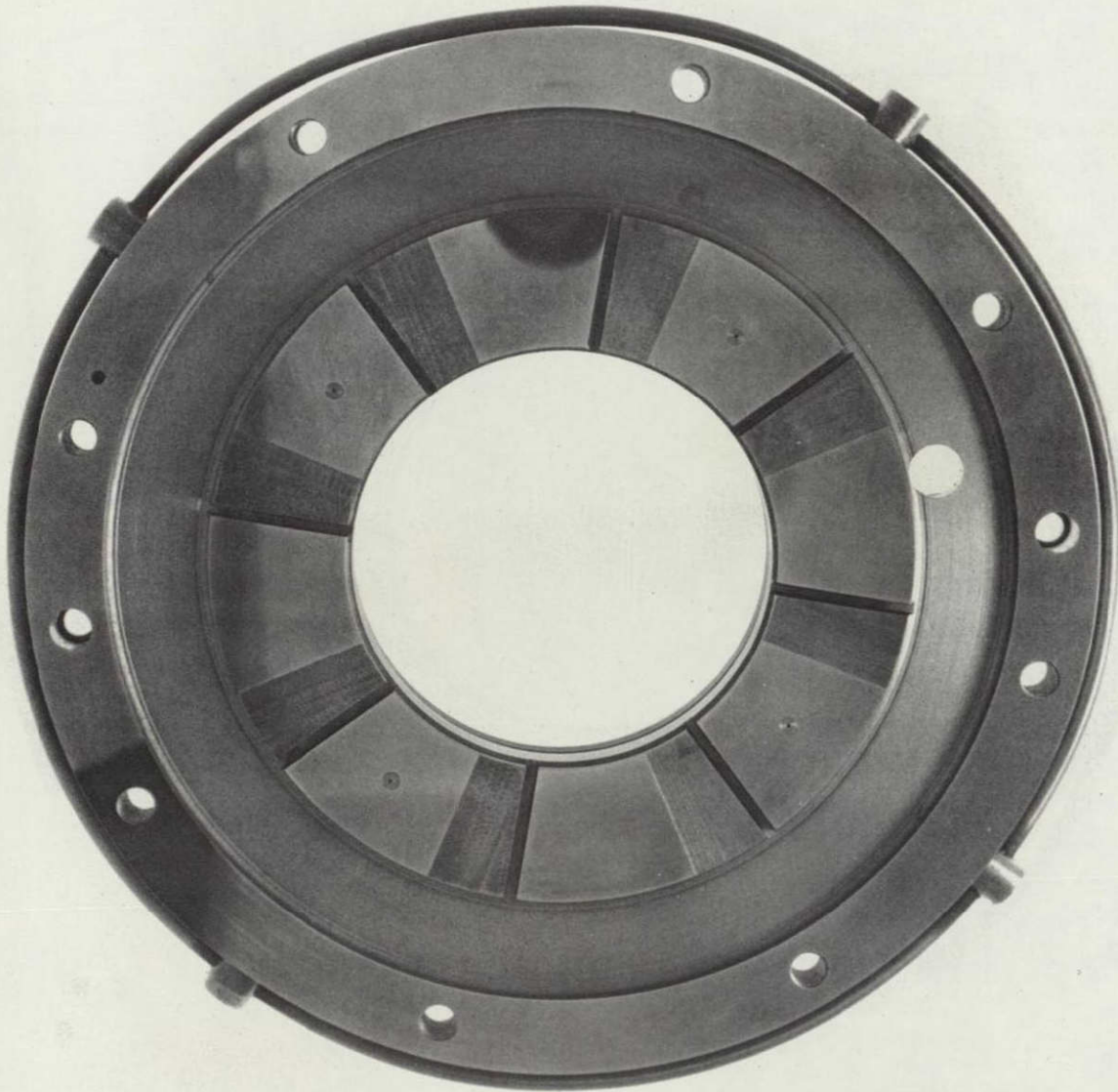


HYDROSTATIC THRUST BEARING STATOR

FIGURE 61



AIRESEARCH MANUFACTURING COMPANY OF ARIZONA
A DIVISION OF THE GARRETT CORPORATION
PHOENIX, ARIZONA

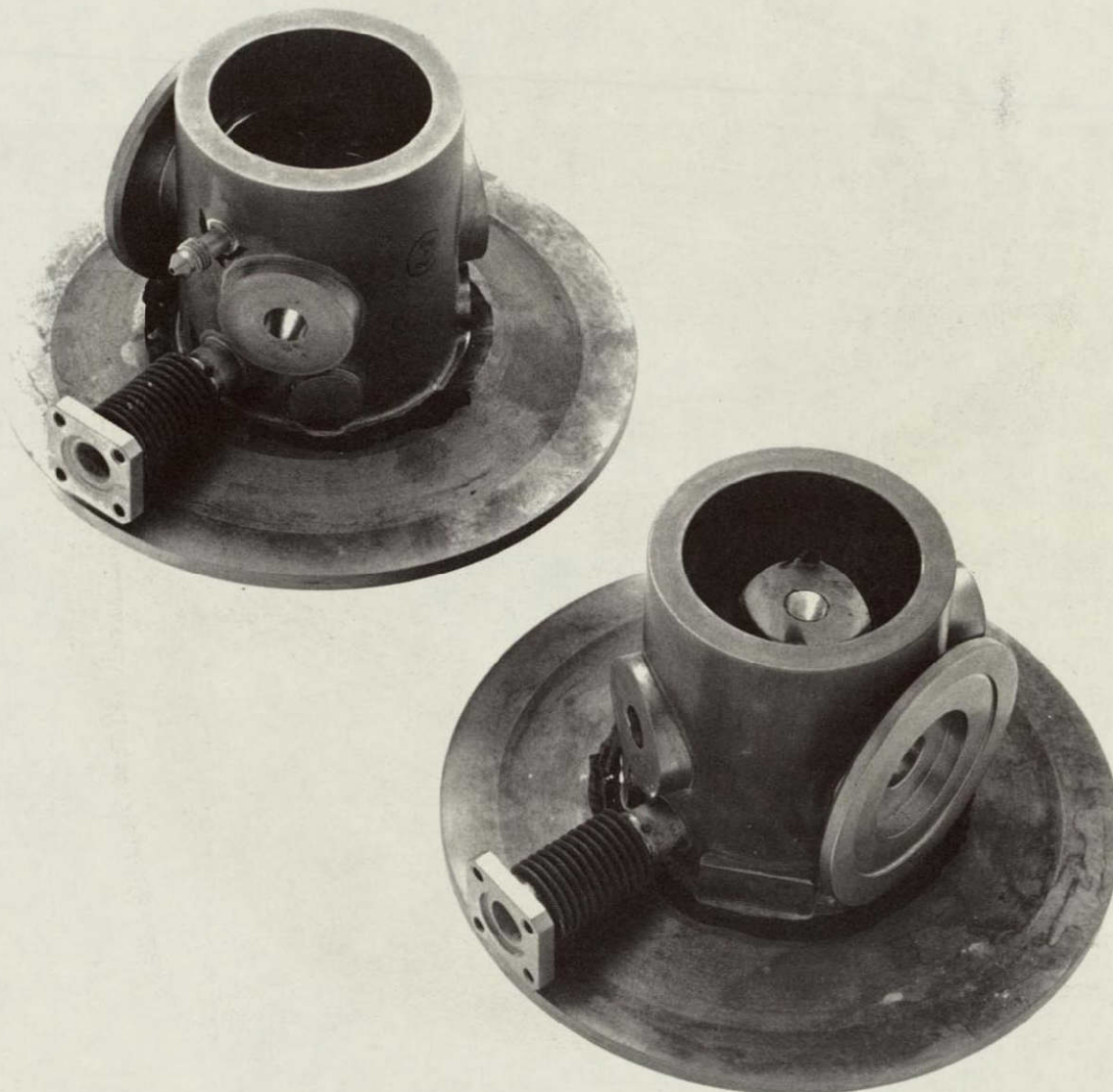


HYDRODYNAMIC THRUST BEARING STATOR

FIGURE 62



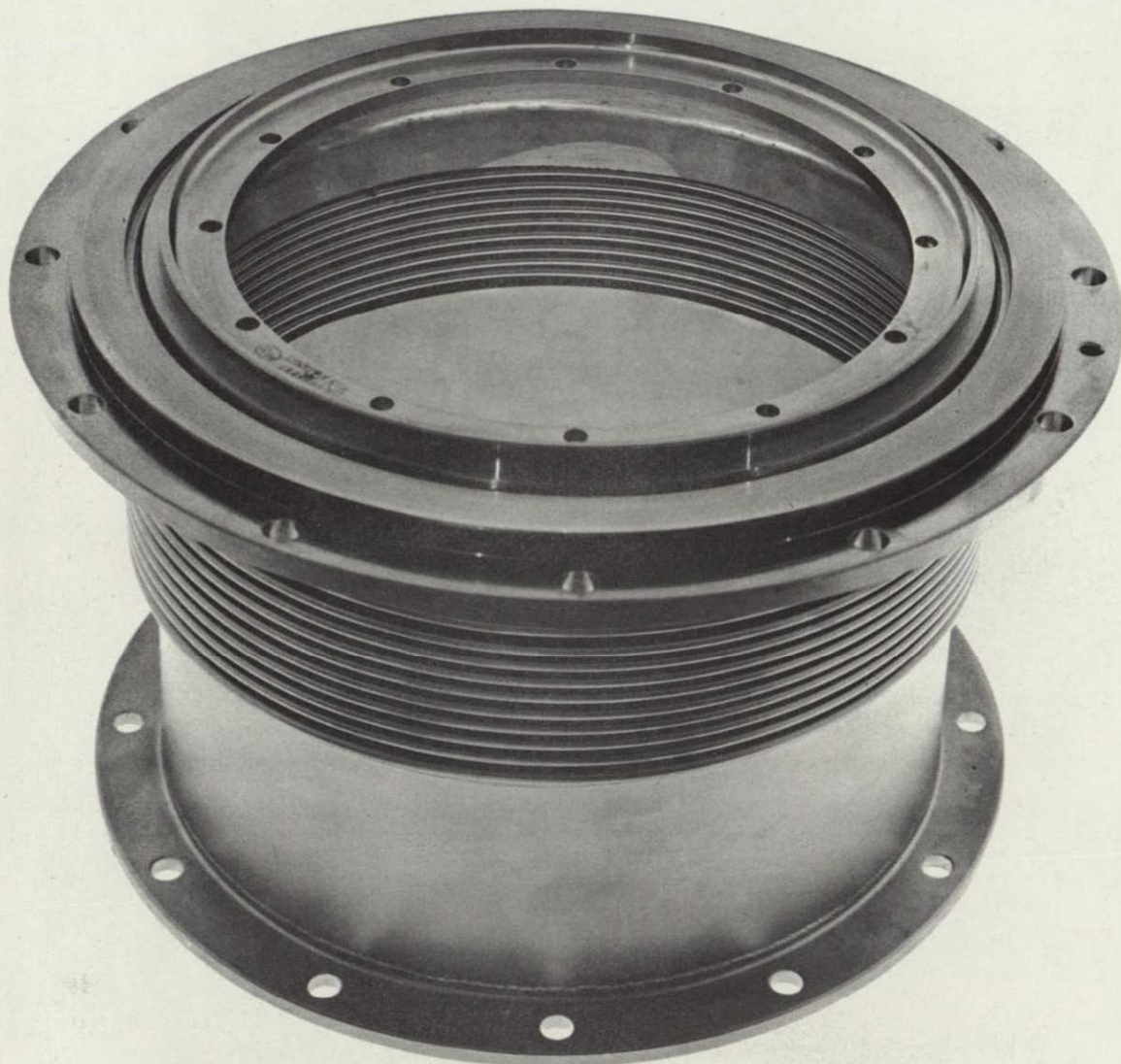
AIRESEARCH MANUFACTURING COMPANY OF ARIZONA
A DIVISION OF THE GARRETT CORPORATION
PHOENIX, ARIZONA



TURBINE AND COMPRESSOR BEARING CARRIERS
FIGURE 63



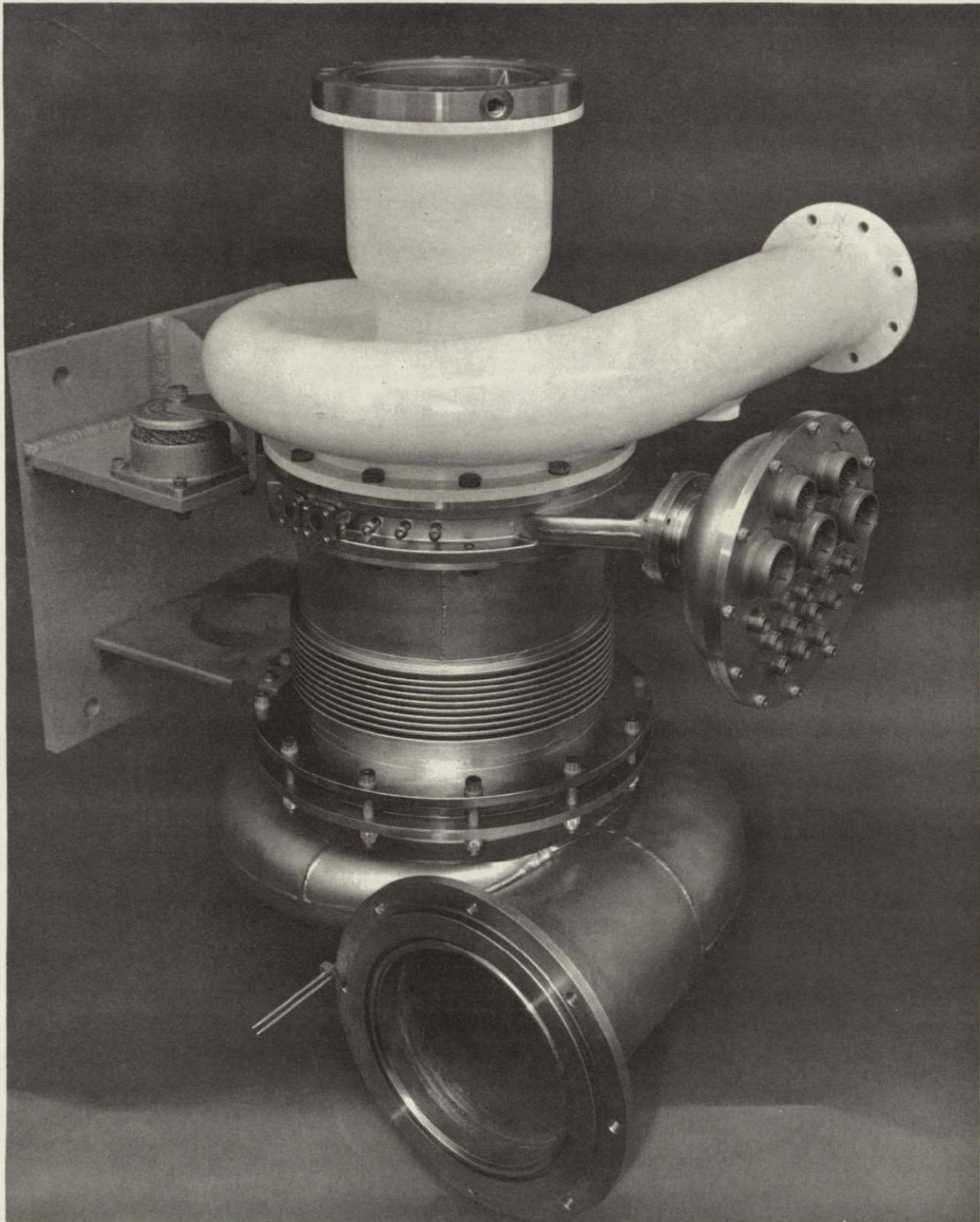
AIRESEARCH MANUFACTURING COMPANY OF ARIZONA
A DIVISION OF THE GARRETT CORPORATION
PHOENIX, ARIZONA



OUTER HOUSING ASSEMBLY
FIGURE 64



AIRESEARCH MANUFACTURING COMPANY OF ARIZONA
A DIVISION OF THE GARRETT CORPORATION
PHOENIX, ARIZONA



NASA BRAYTON-CYCLE GAS GENERATOR

FIGURE 65



AIRESEARCH MANUFACTURING COMPANY OF ARIZONA
A DIVISION OF THE GARRETT CORPORATION
PHOENIX, ARIZONA

TURBINE WHEEL FOR GAS GENERATOR P-A

FORM P2741-F RESEARCH CRITICAL PART INSPECTION AND SERIALIZATION RECORD													
Forging/Casting No.		C/L	Serial Number		O.S. GROWTH DATA				Machined Part No.		C/L		
* 369726-3		B	4X107 (4Q-203)						* 699303-1		N/C		
Oper. Raw Matl.	Date	Insp. Stamp	Oper. Mach. Part	Date	Insp. Stamp	Control Dim/Dia	After G.R.	After O.S.	Z/M After	MRB Disp. After			
										Final	Assy	R&O	
Dim.	2-22-64		Dim.			6.0495		6.0240		USE			
Ultra			Ultra/Mch							RWK			
HT/Stress			HT/Stress							SCP			
Zyglo/Mag	2-22-64		Zyglo/Mag							RTV			
Radiogr.	2-22-64		Pull Test							ITR NUMBER			
Heat No.	RW055		Balance							5464 WSC			
2nd H.T.			Overspeed	MAY 1965 14						23382			
Remarks:			Green Run							*Raw Matl. Mfg. Co.			
			2nd O.S.							*Mach. Part Mfg. No.			
			R.R. No.	IPT # 134789		Part No. Changes							

COMPRESSOR WHEEL FOR GAS GENERATOR P-A

CRITICAL PART INSPECTION AND SERIALIZATION RECORD													
Forging/Casting No.		C/L	Serial Number		O.S. GROWTH DATA				Machined Part No.		C/L		
* 379932-3		J	4L-5545						* 699304-1		"A"		
Oper. Raw Matl.	Date	Insp. Stamp	Oper. Mach. Part	Date	Insp. Stamp	Control Dim/Dia	After G.R.	After O.S.	Z/M After	MRB Disp. After			
										Final	Assy	R&O	
Dim.	10-14-64		Dim.			5.976		5.9762		USE			
Ultra	10-14-64		Ultra/Mch							RWK			
ANN HT/Stress	10-14-64		HT/Stress							SCP			
Zyglo/Mag			Zyglo/Mag							RTV			
Radiogr.			Pull Test							ITR NUMBER			
Heat No.	D-7403		Balance							54060 ENC 8/20			
2nd H.T.			Overspeed	JUN 1965 4									
Remarks: Duplicate card Original one lost 2.358 3.348 S.D. is 2.358 LBA 5/1/65			Green Run							*Raw Matl. Mfg. Co.			
			2nd O.S.							*Mach. Part Mfg. No.			
			R.R. No.	8221		Part No. Changes							

FIGURE 66



AIRESEARCH MANUFACTURING COMPANY OF ARIZONA

A DIVISION OF THE GARRETT CORPORATION
PHOENIX, ARIZONA

TURBINE WHEEL FOR GAS GENERATOR P-B

FORM P2741-F CRITICAL PART INSPECTION AND SERIALIZATION RECORD													
Forging/Casting No.		C/L	Serial Number		O.S. GROWTH DATA			Machined Part No.		C/L			
369726-3		B	4X105 (10-201)					699303-1		N/C			
Oper. Raw Matl.	Date	Insp. Stamp	Oper. Mach. Part	Date	Insp. Stamp	Control Dim/Dia	After G.R.	After O.S.	Z/M After	MRB Disp. After			
										Final	Assy	R&O	
Dim.	2-22-65	(A)	Dim.			6.6790		6.0702		USE			
Ultra			Ultra/Mch							RWK			
HT/Stress			HT/Stress							SCP			
Zygio/Mag	2-22-65	(A)	Zygio/Mag							RTV			
Radiogr.	2-22-65	(A)	Pull Test							ITR NUMBER			
Heat No.	RW055		Balance							23382			
2nd H.T.			Overspeed	MAY 1965	(S)								
Remarks:			Green Run							*Raw Matl. Mfg. Co.			
			2nd O.S.							*Mach. Part Mfg. No.			
			R.R. No.	IPT # 134789		Part No. Changes							

COMPRESSOR WHEEL FOR GAS GENERATOR P-B

CRITICAL PART INSPECTION AND SERIALIZATION RECORD													
Forging/Casting No.		C/L	Serial Number		O.S. GROWTH DATA			Machined Part No.		C/L			
379932-3		J	4L5546					699304-1		H			
Oper. Raw Matl.	Date	Insp. Stamp	Oper. Mach. Part	Date	Insp. Stamp	Control Dim/Dia	After G.R.	After O.S.	Z/M After	MRB Disp. After			
										Final	Assy	R&O	
Dim.	10-65	(G) 1226	Dim.			5.9770		5.9770		USE			
Ultra	10-65	(G) 1226	Ultra/Mch							RWK			
HT/Stress	10-65	(G) 1226	HT/Stress							SCP			
Zygio/Mag			Zygio/Mag							RTV			
Radiogr.			Pull Test							ITR NUMBER			
Heat No.	D-7403		Balance							54061-50			
2nd H.T.			Overspeed	JUN 1965	(S)								
Remarks:			Green Run							*Raw Matl. Mfg. Co.			
			2nd O.S.							*Mach. Part Mfg. No.			
			R.R. No.	8221		Part No. Changes							

FIGURE 67



P-A



AIRESEARCH MANUFACTURING COMPANY OF ARIZONA
A DIVISION OF THE GARRETT CORPORATION
PHOENIX ARIZONA

As can be seen from Figure 68, the first gas generator was run for a total of 5.58 hours during the acceptance test. This test is discussed in Section 5.0.



5.0 ACCEPTANCE TEST

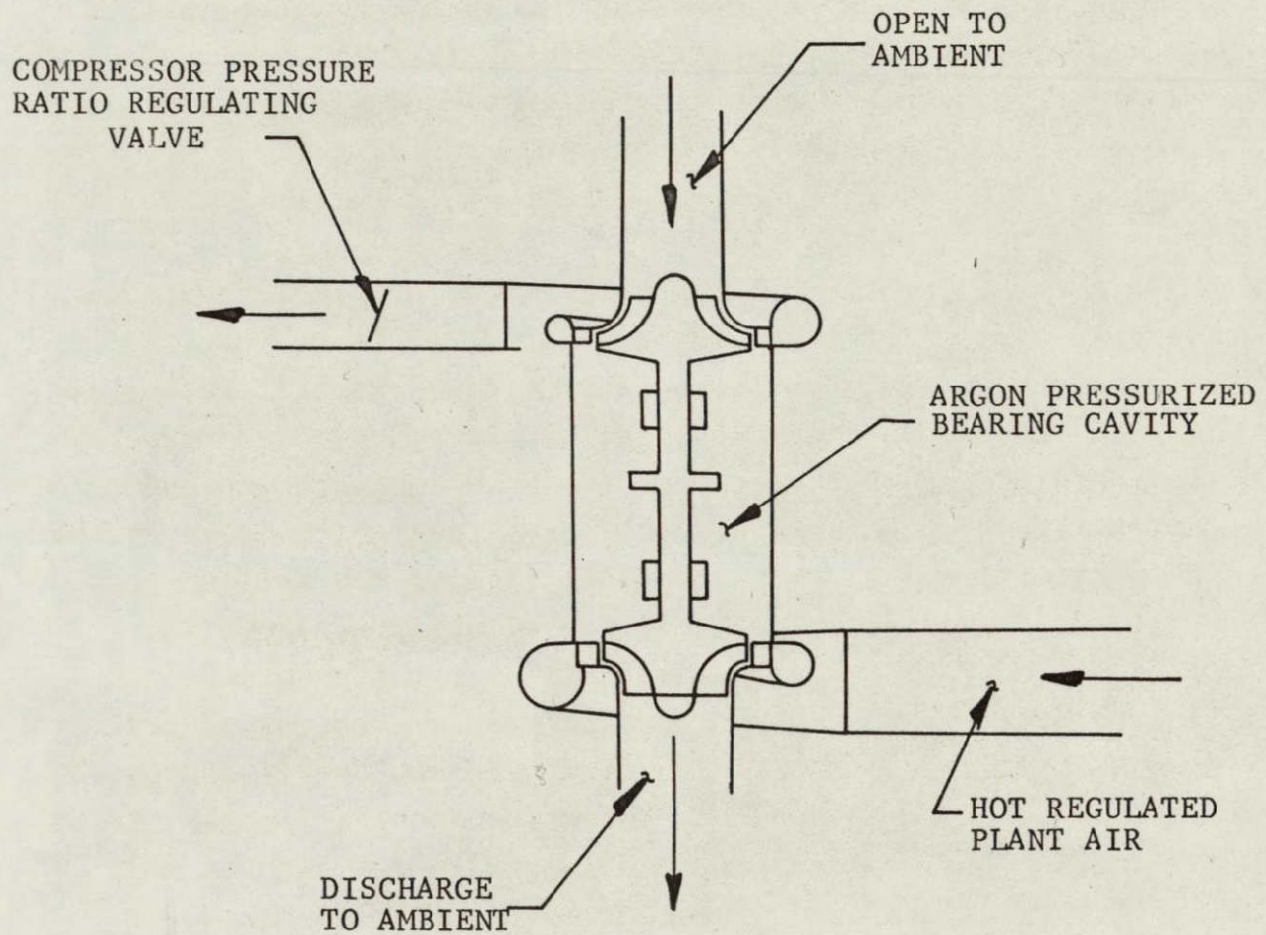
5.1 Test Installation

Operation of the gas generator for the acceptance test consisted of running the unit with plant air supplied to the turbine and with the compressor open to a clean air supply. Bearing and cavity pressurization was accomplished with filtered argon. Figure 69 schematically shows the acceptance test gas generator installation.

With the completion of the thrust bearing/gimbal assembly integration tests utilizing the gas generator frame test rig (see Section 3.5.2.2), the test frame rig was rebuilt with a set of wheels and scrolls to (a) determine the aerodynamic effects of the wheels upon the bearing system when subjected to cold-acceptance-test conditions and (b) check out the cold-acceptance-test installation prior to acceptance testing the delivery gas generator. The initial test-cell installation is shown in Figures 70 and 71. Other than an initial rotating-group balance problem in the unit, the main problem was one of cell piping modifications in order to eliminate extraneous random vibrations fed into the unit by pneumatic ducting forces. A total of 15 hours of test time was logged in the turbine test cell without incident. At the conclusion of these tests, it was considered that the facility was suitable for the cold acceptance test of the NASA delivery gas generator No. 1. Of significant importance was the fact that the behavior of the bearing system, as established in the frame test-rig configuration with dummy rotating masses, was not in any way disturbed by the aerodynamic effect of the added wheels and scrolls.



AIRESEARCH MANUFACTURING COMPANY OF ARIZONA
A DIVISION OF THE GARRETT CORPORATION
PHOENIX, ARIZONA



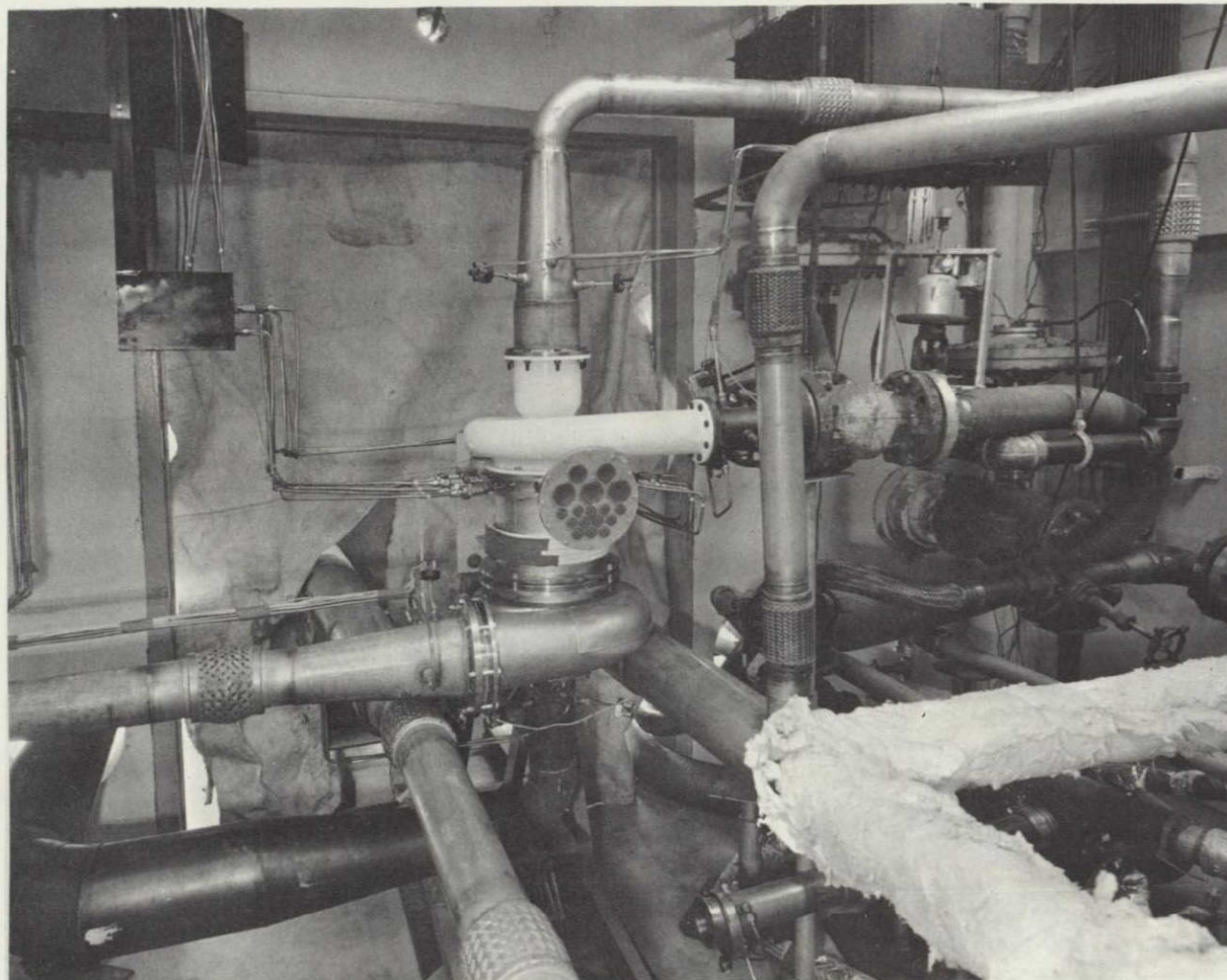
GAS GENERATOR ACCEPTANCE TEST INSTALLATION SCHEMATIC

FIGURE 69



AIRESEARCH MANUFACTURING COMPANY OF ARIZONA

A DIVISION OF THE GARRETT CORPORATION
PHOENIX, ARIZONA

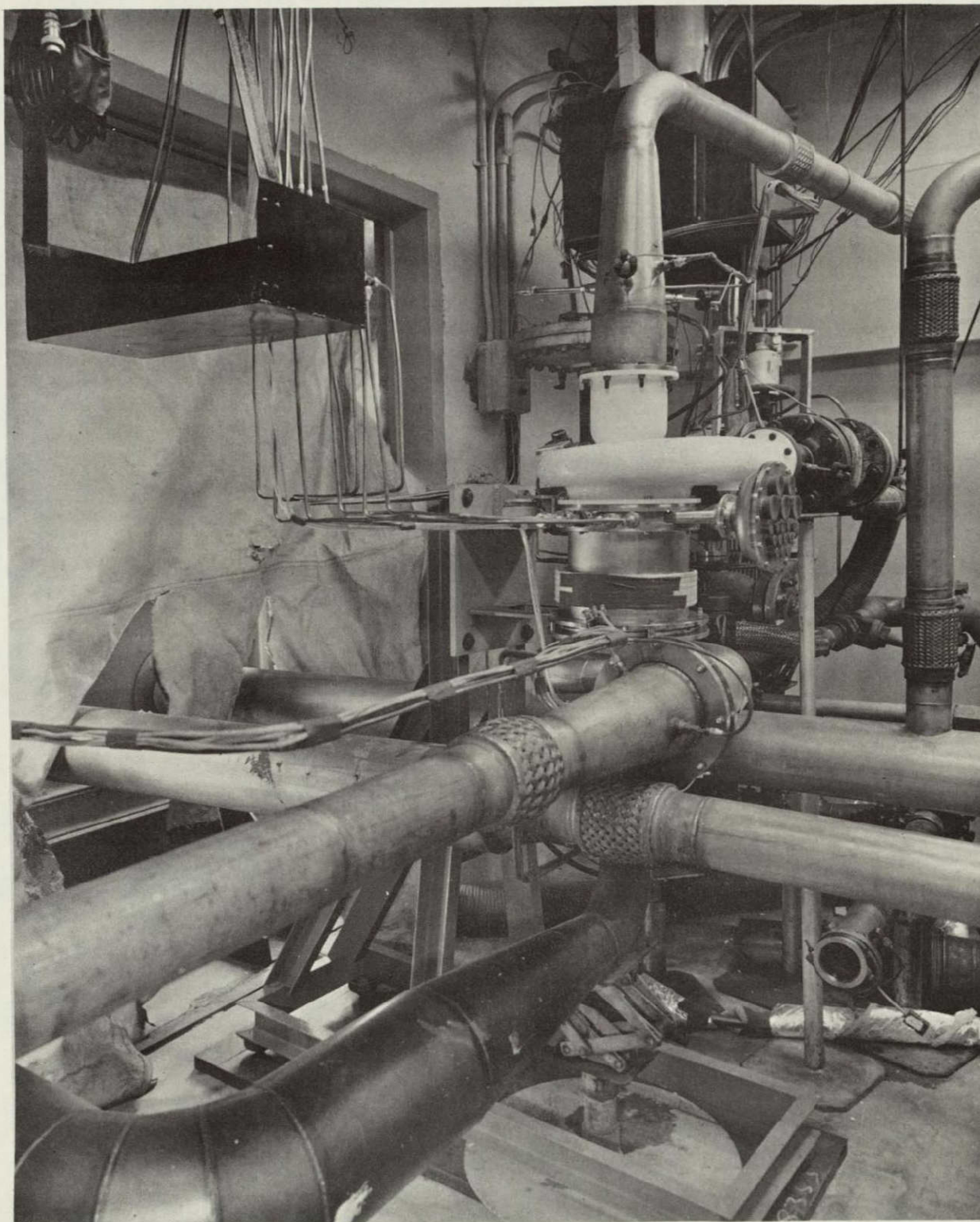


ACCEPTANCE TEST MOCK-UP OF GAS GENERATOR
INSTALLED IN TEST CELL

FIGURE 70



AIRESEARCH MANUFACTURING COMPANY OF ARIZONA
A DIVISION OF THE GARRETT CORPORATION
PHOENIX, ARIZONA



ACCEPTANCE TEST MOCK-UP OF GAS GENERATOR
INSTALLED IN TEST CELL

FIGURE 71



Testing of the gas generator in the turbine test cell indicated that the net aerodynamic thrust of the two wheels was in the reverse-thrust direction and that at 20,000 rpm the addition of temperature to the turbine inlet air in the range of 450 to 500°F allowed speed to be increased to 100 percent without subjecting the unit to excessive reverse-thrust loads. At 100 percent design speed, the resultant thrust load was of the order of 20 pounds in the reverse direction, which indicated that a balance of aerodynamic forces had been achieved. Test-plan requirements for the acceptance test were thus established.

5.2 Assembly of Gas Generator, First Shipping Unit

The first shipping unit, Serial No. P-A, Outline 699300, was assembled, and the assembly data on this unit was collected and documented in AiResearch Report APS-5174-R and forwarded to the NASA when the unit was shipped. The data relevant to the assembly is as follows:

- Shaft-to-shoe geometric clearances.

- Bearing mount shimming log, including bearing preload settings, axial center shift, and seal concentricities.

- Turbine and compressor axial shroud clearance data.

- Capacitance probe and speed pickup shimming log.

- Capacitance probe calibrations (6)

- Gas generator thrust-mount load calibration.

- Compressor and turbine journal bearing flexible-mount diaphragm calibrations (2).

- Instrumentation pin call-out data.

- Instrumentation location schematic



For safety considerations, the axial shroud clearances were increased over the design requirements to allow for test experience to be gained on the unit prior to extracting final efficiency figures. The data presented will allow the NASA to remove material from the shims in successive stages in achieving the end objective during closed-loop testing.

5.3 Initial Test and Calibration of First Shipping Unit

The shipping unit, after basic buildup, was equipped with the dummy masses and tie-bolt from the frame test rig. The rotating group was then subjected to gas bearing evaluation tests in order to determine unit performance and to check out the instrumentation, particularly the proximity probes and certain specific thermocouples. Checkout included operation of the bearing journals and thrust faces hydrostatically, with the reverse thrust bearing being calibrated for film thickness versus load. The normal thrust bearing was also calibrated in the same manner when in the hydrostatic regime. A single-point calibration was also established when the normal thrust bearing was run hydrodynamically. These calibrations have been reported in AiResearch Report APS-5174-R. The journal bearings were also operated in the hydrostatic regime. Additional thrust-transfer checks were performed in both directions with hydrostatic operation, between normal and reverse-thrust faces at 35,000, 30,000, 25,000, and 20,000 rpm. Overspeed was also checked in the hydrostatic mode to 42,000 rpm. All of the tests conducted in the bearing test cell were performed without any problems occurring. The unit proved to have even better performance than had been demonstrated previously on the company-sponsored test rig, particularly with respect to shaft balance and thrust stator face operating clearances.



5.4 Acceptance Test of First Shipping Unit

The test-plan procedure had been adequately established by the prior test series undertaken with the modified frame test rig. The installation configuration in the test cell was similar to Figures 70 and 71. Figures 72 and 73 show the test instrumentation setup of the Wayne-Kerr proximity meters in relation to the test unit and the visual recording data outputs, respectively.

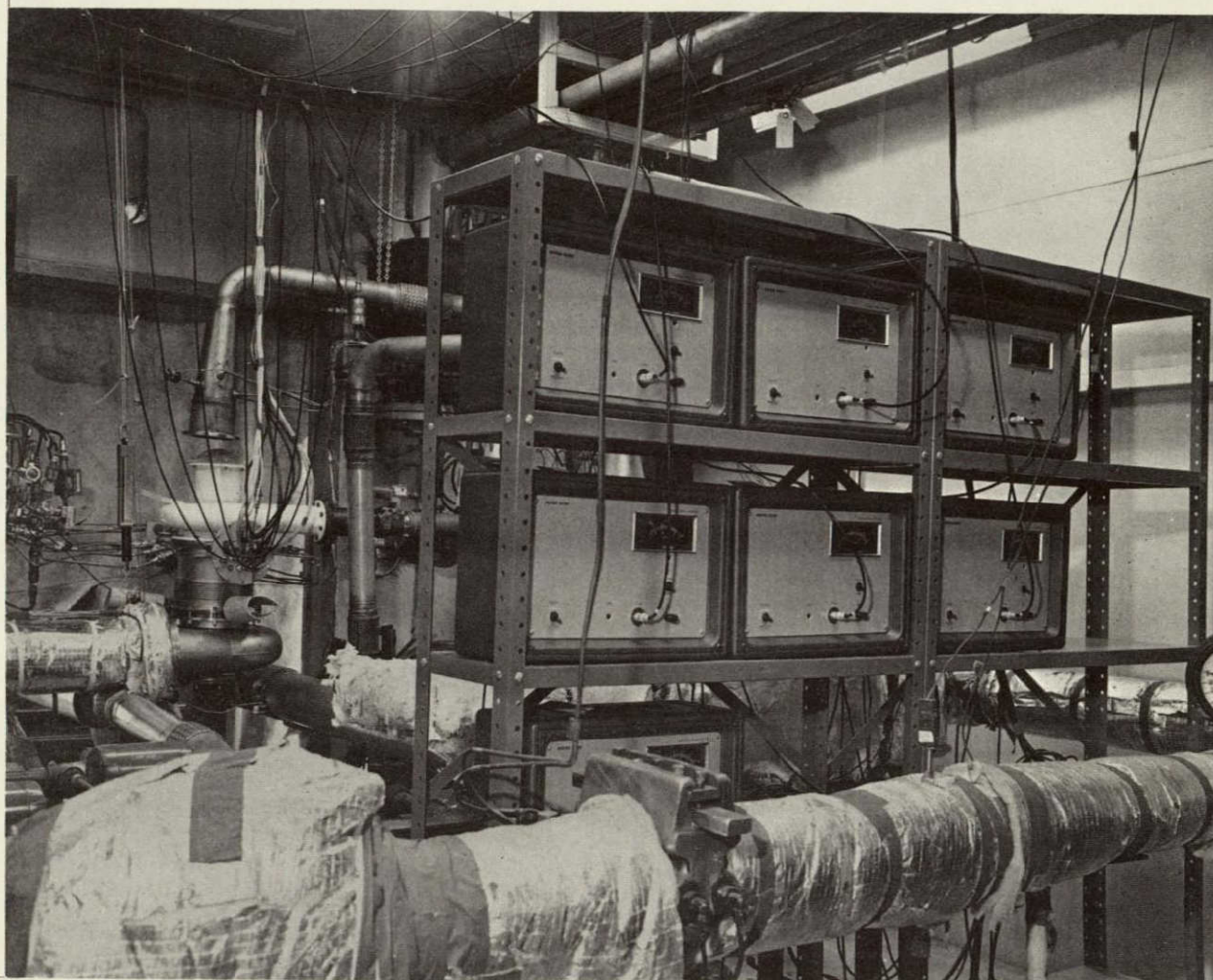
After installation, the unit was subjected to a test run to full speed, hydrostatically, to determine that the rotational balance of the shaft components was acceptable. The balance was proven to be satisfactory.

The final instrumentation hook-up for recording purposes was arranged as follows:

Tape Recorder:	Seven channels, to record the four orthogonal and two thrust probes, with speed and voice combined on one channel.
Oscillograph Recorder:	20 channels, to record 16 thermocouples, 3 strain gauges, and speed.
Visual Readout:	The remaining thermocouples chosen for safety reasons. Shaft and thrust surface proximity probes, speed, temperature, and strain-gauge output on the thrust mount through an X-Y plotter.



AIRESEARCH MANUFACTURING COMPANY OF ARIZONA
A DIVISION OF THE GARRETT CORPORATION
PHOENIX, ARIZONA



WAYNE-KERR PROXIMITY METERS
NASA GAS GENERATOR ACCEPTANCE TEST

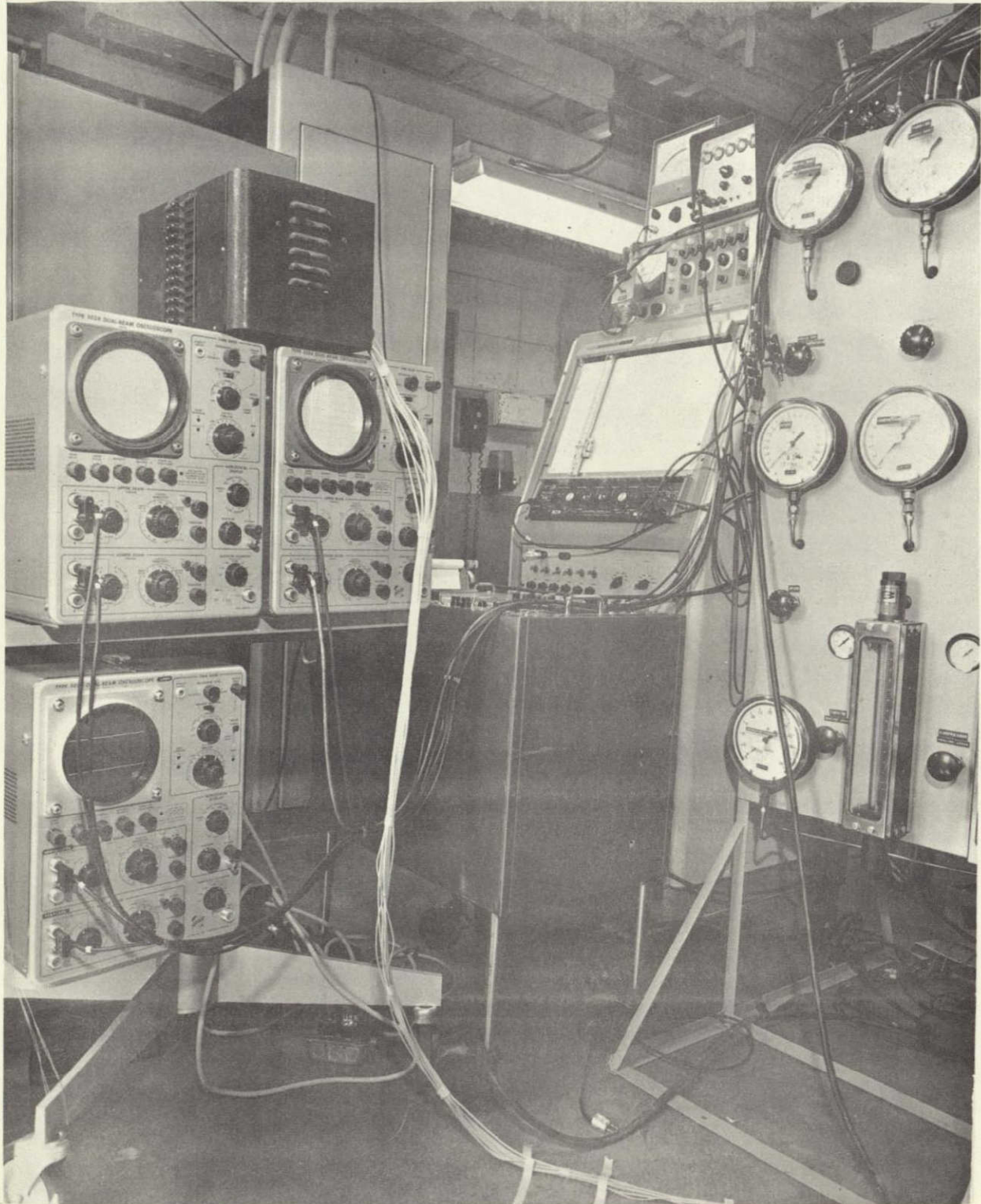
FIGURE 72



AIRESEARCH MANUFACTURING COMPANY

A DIVISION OF THE GARRETT CORPORATION

PHOENIX, ARIZONA



GAS BEARING CONTROL PANEL AND INSTRUMENTATION
NASA GAS GENERATOR ACCEPTANCE TEST

FIGURE 73



AIRESEARCH MANUFACTURING COMPANY OF ARIZONA
A DIVISION OF THE GARRETT CORPORATION
PHOENIX, ARIZONA

The acceptance test was run in accordance with the test plan and successfully demonstrated the requirements of the contract. This test was witnessed by the NASA representatives, and the unit was accepted.

Although performance demonstration was not required, measurements were required to establish steady-state operation and control during tests. Figure 74 shows the test-cell log sheet for the acceptance test on the Serial No. P-A gas generator, and Figures 75 and 76 show the data log sheets for the acceptance test. Figure 77 shows the tape and oscillograph log sheets. Copies of the tape record and the oscillograph were shipped with the gas generator.

The second gas generator was shipped as components, at the request of the NASA, and will be assembled at the NASA.

QUALIFICATION TEST LOG

E.W.O. No. 2409-27136-16-1006 Date 9-9-65 Test Cell or Station No. D-107
Assembly No. 699301 Model No. CCGG-20-1 Unit Serial No. P-A
Development Engineer B HEATH Technician BLENDEIC Grp. Ldr. TIMMEIC
Test Type DEV. Test Schedule APS 5173-R Modification -

TIME		Event	O.C.
START	STOP		
		DRYS; CALIBRATED ALL INSTRUMENTATION	
		RUN #1 TAPE #800 (C-1400)	
10:20		TOOK STATIC DATA COLL FA	
10:25		BEGIN RUN, ACCELL. TO 20000 RPM & INCR TURB. INLET TO 400°F	(52)
10:30		ACCELL. TO 32500 RPM	
10:35		DATA COLL. BB	
10:42		REMOVED JOURN. BEG. H.S. ARGON & INCR. TURB. INLET TO 475°F	(53)
10:45		DATA COLL. CC	
11:15		DATA COLL. DD	
11:45		DATA COLL. EE	
12:15		DATA COLL. FF	
12:45		DATA COLL. GG	
13:15		DATA COLL. HH	
13:45		DATA COLL. II	
14:15		DATA COLL. JJ	
14:45		DATA COLL. KK	
15:15		DATA COLL. LL	
15:45		DATA COLL. MM	
		RETURNED HYDROSTATIC ARGON TO JOURN. BEG. (GO PSIA)	
15:50		REDUCED SPD TO 20000 RPM & TOOK DATA (TAPE) EVERY TWO MIN. FOR TEN MIN.	
15:55		INCR. COOLING ARGON FLOW TO 60 SCFH ON TURB. & COMP.	(54)
16:00		SHUT DOWN	
		REMOVED UNIT FROM CELL	

SUMMARY: Total Running Time 5 hrs. 35 min.
Total Manual Starts
Total Automatic Starts

Ref. Data Page

Engineering

R. B. Blendeic



Reproduced from
best available copy.



FORM T-700

[illegible]

NASA GAS GENERATOR ACCEPTANCE TEST LOG SHEET

FIGURE 76
124

Unit Type: NASA GAS GEN EWO: 6701 Page No.: 1
Test Location: CELL D.107 EWO: 3409-27136-16-1002 Recorder No.: 0

Record Number	Date	Time	Tech	Event	Remarks	Unit S/N	Spot Checked By
	9-9		RAS	NASA SHIPPING UNIT		P-A	
00390		1020			RUN UP TO 20,000 RPM		(62)
00391		1030		DP NO. 1	RUN UP TO 38,500 RPM		2
00392		1042		DP NO. 2	RUN HYDRODYNAMIC ON JOURNALS		(22)
00393		1115		DP NO. 3	HYDRODYNAMIC AT 38,500 RPM		(22)
00394		1135		DP NO. 4	HYDRODYNAMIC AT 38,500 RPM		
00395		1215		DP NO. 5	HYDRODYNAMIC AT 38,500 RPM		
00396		1245		DP NO. 6	HYDRODYNAMIC AT 38,500 RPM		
00397		1315		DP NO. 7	HYDRODYNAMIC AT 38,500 RPM		
00398		1345		DP NO. 8	HYDRODYNAMIC AT 38,500 RPM		
00399		1415		DP NO. 9	HYDRODYNAMIC AT 38,500 RPM		
00400		1445		DP NO. 10	HYDRODYNAMIC AT 38,500 RPM		
00401		1515		DP NO. 11	HYDRODYNAMIC AT 38,500 RPM		(62)
00402		1535		DP NO. 12	HYDRODYNAMIC AT 38,500 RPM		
00403		1547		DP NO. 13	ADDED HYDROSTATIC ARGON AND REDUCE SPEED TO 20,000 RPM		(62)
00404		1555		DP NO. 14	STEADY STATE AT 20,000 RPM		
00405		1557		DP NO. 15	STEADY STATE AT 20,000 RPM		
00406		1559		DP NO. 16	STEADY STATE AT 20,000 RPM		
00407		1601		DP NO. 17	STEADY STATE AT 20,000 RPM		
00408		1602			ROLL DOWN		(62)

EX1406

OSCILLOGRAPH RECORDING DATA

Unit Type: NASA GAS GEN. Test Location: D-107 Technician: Shumaker Test Speed: 25 in./sec. L Calve Volts: 7.0 Cart No: 6
Job Title: ACCEPTANCE TEST XWO: 6701 Engineer: B. A. M. Paper Type: 12" W-3 R Calve Volts: 7.5 Recorder No: 6
Date: 9-9-65 XWO: 3809-27136-16-1002 Requested By: Timing Lines 1 per Sec. Timing Volts: 0 Record No: 00390 To: 00408
UNIT ASSEMBLY 699300, SERIAL NO. J.P.A.

Test No.	Parameter	Console Chan	Calibration	Galvanometer		Cal. No.	Transducer		Accuracy			Trans. Cable	Patch No.	Remarks	
				Ref. Type	Damping		Point	Type	S/N	Range	Ind.				Full Spec
1	SPEED	TACH	10,000 RPM/IN	0.0											
2	THERMOCOUPLE NO. 1	T-1	200°F/IN 0°F AT	1.0											
3	THERMOCOUPLE NO. 2	T-2	200°F/IN 0°F AT	1.0											
4	THERMOCOUPLE NO. 4	T-3	200°F/IN 0°F AT	1.0											
5	THERMOCOUPLE NO. 6	T-4	200°F/IN 0°F AT	2.0											
6	THERMOCOUPLE NO. 7	T-5	200°F/IN 0°F AT	2.0											
7	THERMOCOUPLE NO. 10	T-6	200°F/IN 0°F AT	2.0											
8	THERMOCOUPLE NO. 11	T-7	200°F/IN 0°F AT	3.0											
9	THERMOCOUPLE NO. 13	T-8	200°F/IN 0°F AT	3.0											
10	THERMOCOUPLE NO. 14	T-9	200°F/IN 0°F AT	3.0											
11	THERMOCOUPLE NO. 15	T-10	200°F/IN 0°F AT	4.0											
12	THERMOCOUPLE NO. 16	T-11	200°F/IN 0°F AT	4.0											
13	THERMOCOUPLE NO. 18	T-12	200°F/IN 0°F AT	4.0											
14	THERMOCOUPLE NO. 19	T-1A	200°F/IN 0°F AT	5.0											
15	THERMOCOUPLE NO. 20	T-2A	200°F/IN 0°F AT	5.0											
16	THERMOCOUPLE NO. 21	T-3A	200°F/IN 0°F AT	5.0											
17	THERMOCOUPLE NO. 22	T-4A	200°F/IN 0°F AT	6.0											
18	TURBINE BRG. LOAD	ATT NO. 1	10 LBS/IN	6.0		345"	14 LB	56K	INCREASING LOAD						ZERO W/NORMAL PRELOAD
19	COMER BRG. LOAD	ATT NO. 2	10 LBS/IN	7.0		335"	8 ALB	56K	INCREASING LOAD						ZERO W/NORMAL PRELOAD
20	CRUST BRG. LOAD	ATT NO. 3	200"/IN	8.0		195"		100K	CAL IN HYDRODYNAMIC DIRECTION						ZERO-ROTOR WT.
(22)															

125



APPENDIX I

Glossary

The following is a list of terms used throughout the report:

- g = conversion factor = 32.2 ft lb per lb sec².
 \dot{m} = molal gas flow rate, lbs mol per sec.
 r_c = compressor pressure ratio
 r_{t_1} = gas-generator turbine pressure ratio
 r_{t_2} = power-turbine pressure ratio
 C = difference between bearing shoe radius and journal radius, inches
 D_c = compressor-wheel diameter, inches
 D_{t_1} = gas-generator turbine-wheel diameter, inches
 E_R = recuperator effectiveness
 L = bearing axial length, inches
 M = molecular weight, lbs per lb mol
 N_1 = gas-generator shaft speed, rpm
 N_2 = power-turbine shaft speed, rpm
 N_{S_c} = compressor shaft speed
 $N_{S_{t_1}}$ = gas-generator turbine specific speed
 $N_{S_{t_2}}$ = power-turbine specific speed
 P_1 = compressor inlet pressure, lbs per sq ft
 P_3 = gas-generator turbine inlet pressure, lbs per sq ft
 R = bearing journal radius, inches
 R_1 = universal gas constant = 1545 ft-lbs per lb-mol °R



AIRESEARCH MANUFACTURING COMPANY OF ARIZONA
A DIVISION OF THE GARRETT CORPORATION
PHOENIX, ARIZONA

T_1 = compressor inlet temperature, $^{\circ}\text{R}$

T_3 = gas-generator turbine-inlet temperature, $^{\circ}\text{R}$

W = mass flow rate, lbs per sec

β = $\frac{\text{turbine pressure ratio}}{\text{compressor pressure ratio}} = \frac{r_{t1} \times r_{t2}}{r_c}$

δ = inlet pressure, psia/14.7 psia

γ = ratio of gas specific heats = 1.667 for monatomic gases

θ_1 = $\gamma - 1/\gamma$

θ = inlet temperature, $^{\circ}\text{R}/518.7^{\circ}\text{R}$

η_c = compressor adiabatic efficiency

η_{cy} = $\frac{\text{power-turbine shaft power output}}{\text{gas-cycle input rate}}$

η_{t1} = gas-generator turbine adiabatic efficiency

η_{t2} = power-turbine adiabatic efficiency

NASA CONTRACTOR REPORT

DESIGN AND FABRICATION OF A
HIGH PERFORMANCE BRAYTON CYCLE
RADIAL-FLOW GAS GENERATOR

AIRESEARCH MANUFACTURING COMPANY OF ARIZONA
A DIVISION OF THE GARRETT CORPORATION

ABSTRACT

The program described herein consisted of analyses, design, gas bearing development and fabrication of a radial-flow gas generator (turbocompressor) for performance evaluation by the NASA. The respective turbine and compressor components, identical to those in the gas generator, were delivered in separate research packages utilizing oil-lubricated ball bearings for detail aerodynamic performance investigations. A gas generator was delivered that operated satisfactorily on its gas journal and thrust bearings at 38,500 rpm design speed at 500°F turbine inlet temperature.